BRÜEL & KJÆR NÆRUM · DENMARK

BRÜEL & KJÆR application notes

Field Balancing Equipment Helps The Development Engineer

The crankshaft of an internal combustion engine possesses severe design problems. Its geometric form is irregular, and it is subjected to a variety of cyclic forces, moments, and torsional influences. Subsequent development work is essential to ensure that manufacturing tolerances, crankshaft loading, and support stiffness are adequate so that transmission of these influences does not affect adversely the reliability or useability of the complete engine in service. The trend for increasing specific power output combined with reduced weight has fostered the introduction of successive generations of engines of high rotational speed yet lighter construction. Thus balance quality, which has mechanical, material, and fabrication implications, as well as environmental, noise, and vibration effects, is an important consideration in modern engine design.

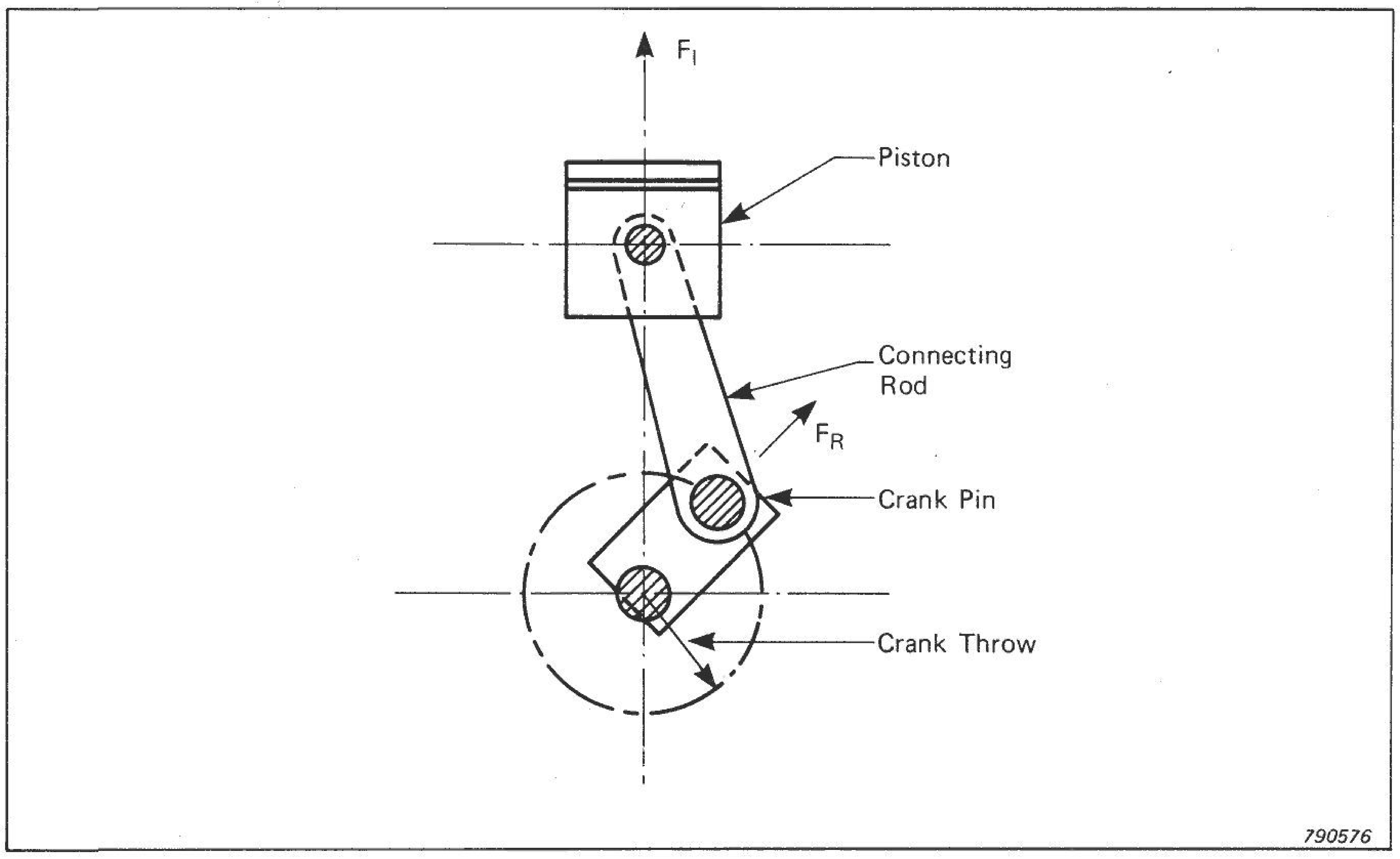

Fig.1. Basic single-cylinder layout

For the basic single-cylinder configuration shown in Fig.1, the radial forces on the crank can be considered in two parts:

Centrifugal forces due to the rotation of masses associated with the crankpin - F_R.

Inertia forces due to the motion of reciprocating masses associated with the piston - F_I.

In addition to the "primary" reciprocating effect of the piston motion, the more complex motion of the link connecting rod gives rise to "secondary" forces at twice the speed of rotation. Furthermore, couples are produced when forces act in different planes along the shaft axis, tending to rock the engine about some axis perpendicular to the cylinder plane. This effect is shown for a twin-cylinder in-line configuration in Fig.2.

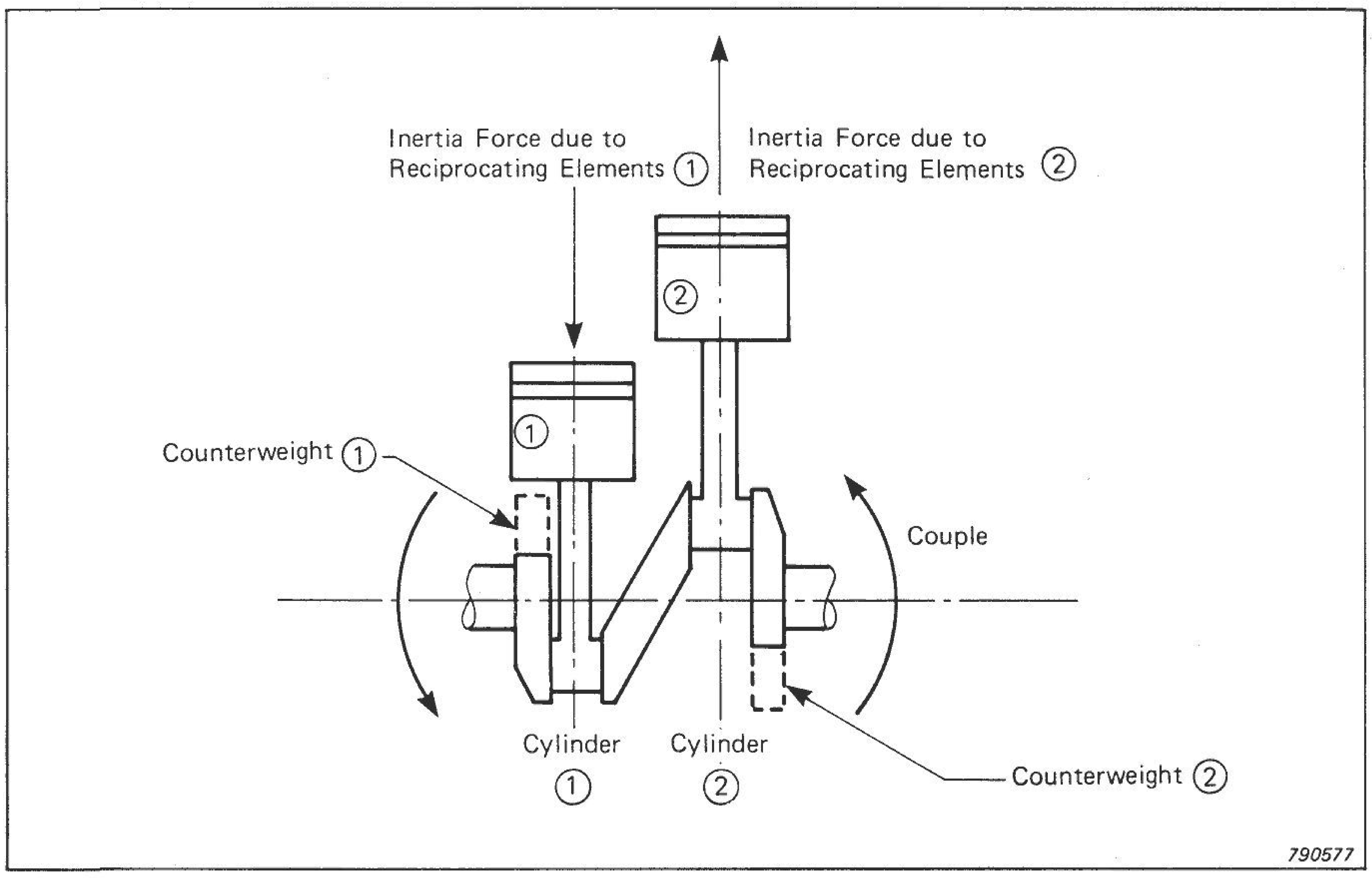

Fig.2. Two-cylinder in-line layout

The connecting rod mass can be separated conveniently into a rotating component, that part associated with the crankpin, and a reciprocating component, that part associated with the piston. To do this the rod is laid horizontal, with the big - and small - ends each supported on their respective balances. To achieve balance in the rotating crankshaft, suitable counterweights must be fitted. There is no difficulty in arranging to balance the rotating mass, but how should reciprocating masses be treated? If the crankshaft counterweights are designed to compensate for these forces fully, an extra unbalance force results at approximately mid-stroke as shown in Fig.3.

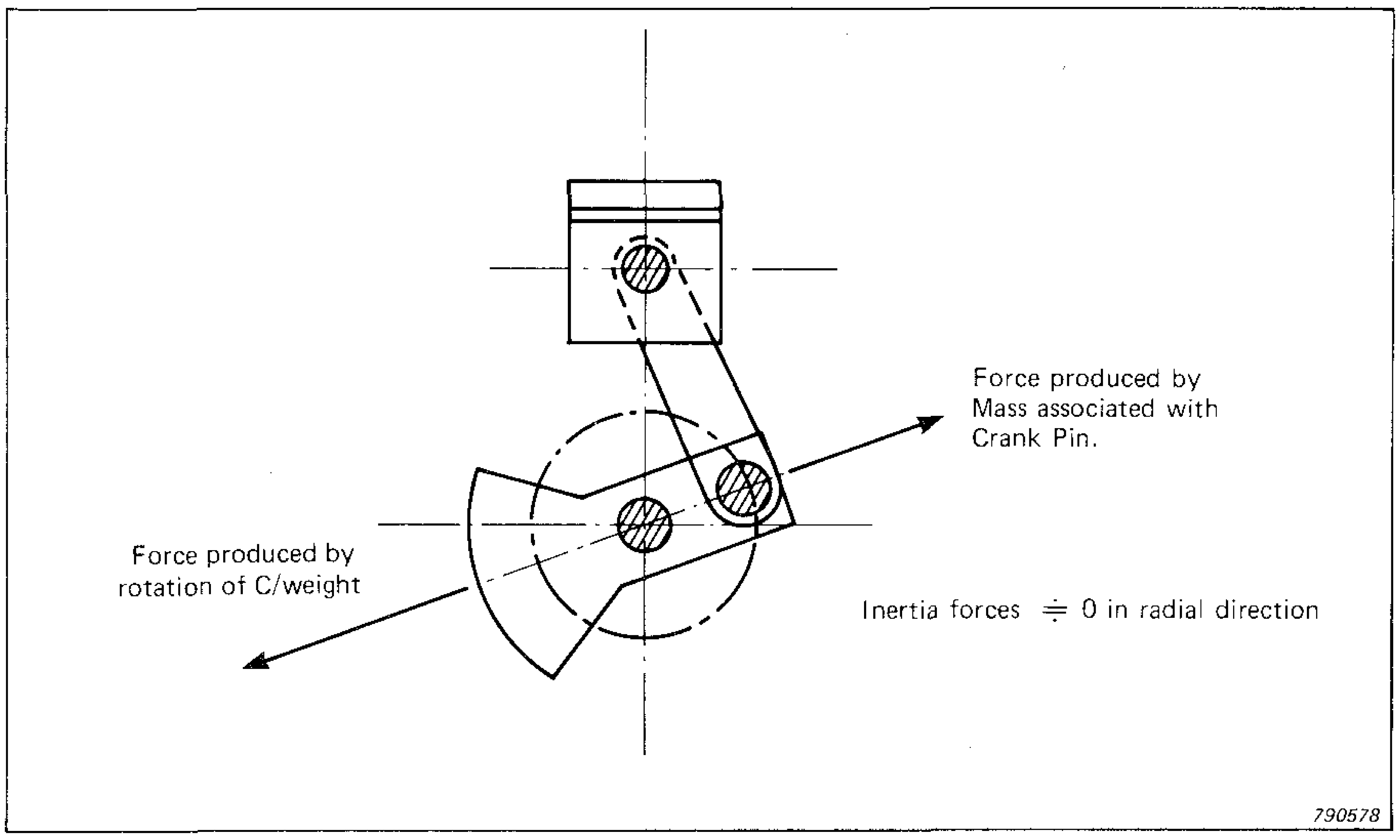

Fig.3. Single-cylinder layout with counterweight

As a compromise solution it is common to use a "balance factor" which is typically 40 - 80% of these inertia forces: the value selected for this factor is determined by the design speed, engine geometry, and structural rigidity of the crankcase. Thus the effective unbalance to be corrected by the counterbalance weight is:

$$F_{tot} = F_R + k\,F_I,$$

where k = balance factor ($0 < k < 1$).

The use of such counterweights on the twin-cylinder configuration, also counteracts the couple effects shown in Fig.2. At best this is only a compromise solution, as the situation becomes further complicated in the assembled engine: then, the combustion forces, connected shafts, gears, and rotating assemblies will combine with the effects of assembly tolerances to influence further the vibration transmitted through the bearings to the crankcase. Actually, complete balance of primary and secondary forces is achieved only by using counter-rotating balance shafts, or a more complex multi-cylinder configuration. The straight 6-cylinder engine is an example of a layout where both primary and secondary effects can be balanced.

This note describes how field balancing equipment was used to investigate the unbalance condition of a Volvo-Penta 15 hp, twin-cylinder, outboard motor operating in a test tank. Two Piezoelectric Accelerometers Type 4371 were attached using studs to the crankcase at the main bearing housings, to measure radial vibration. A photoelectric tachometer probe MM 0012 was mounted on a bracket to trigger from a piece of black tape on the motor flywheel. The two crankshaft counterweights had been drilled axially with holes 18 mm dia: in this way, different unbalance conditions could be investigated by inserting steel plugs into the holes. The overall arrangement with connections to the Portable Balancing Set Type 9500 is shown in Fig.4. Measurements were made at 2800 RPM and 4800 RPM respectively to investigate the state of dynamic balance. The procedure was as follows:

1. Both counterbalance holes empty: run engine up to speed, and measure the unbalance component of vibration in each measuring plane.

2. Strip engine, and fit steel plug to counterbalance 1. Reassemble engine and repeat measurements.

3. Strip engine, and transfer plug to counterbalance 2. Reassemble and repeat measurements.

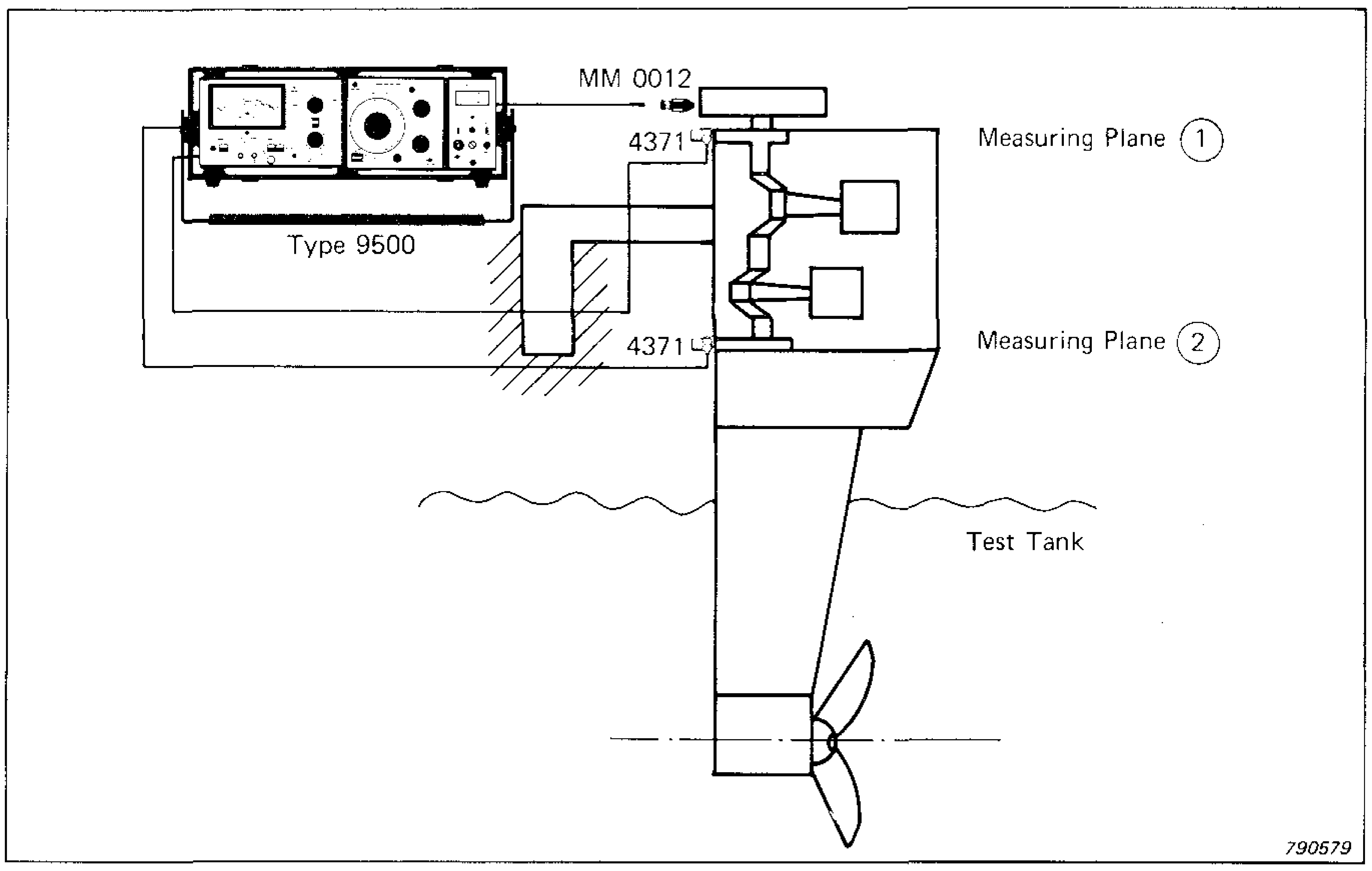

Fig.4. Schematic diagram of test layout: engine in tank

The fact that the speed of such a motor varies slightly, and that the engine itself is subject to non-steady influences such as combustion variation, gave slight fluctuations in the phase readings. However, as they were cyclic in nature it was possible to take representative readings. The measured values for each test speed were fed into the standard B & K DYNBAL program on a Texas TI-59 pocket calculator. The calculated results then showed where the maximum synchronous radial forces occurred, and their relationship to mass placed in the crankshaft counterbalance planes.

For the study of crankshaft balance, it is convenient to separate the mass correction values into their vertical, and horizontal components: that is, the radial component acting through the crankpin, and the component acting at right angles respectively. The convention adopted is shown in Fig.5.

The results have been shown in this form in Table 2, for the two test speeds.

These results can be used to make certain deductions about the synchronous (unbalance) forces acting in the engine.

Considering the vertical component, it may be seen that at 2800 RPM, the counterweights should be further lightened to achieve the optimum balance. In practice, this would mean that a lower balance factor for the reciprocating masses would be specified for production balancing during manufacture. At 4800 RPM the mass addition required has an average value of 1,0, indicating that the balance factor of the test engine with the plugs refitted is quite correct for operation at this speed. At the test speeds, the difference between corrections required to top and bottom counterweights indicates that on the top counterweight an average of 0,76, or an estimated 33 g, is required than on the lower. This is an indication of the mass variations between the two cylinders due to production tolerances of components such as crankpins, connecting rods, and pistons.

	Measuring Position 1 (Top)	Measuring Position 2 (Bottom)
Initial Run	$V_{10} = 46 \angle 92^\circ$	$V_{20} = 32 \angle 92^\circ$
Plug in C/weight 1	$V_{11} = 27 \angle 224^\circ$	$V_{21} = 25 \angle 243^\circ$
Plug in C/weight 2	$V_{12} = 100 \angle 74^\circ$	$V_{22} = 85 \angle 76^\circ$
Plug Mass	1,0*	1,0*
Calculated Correction Mass (relative to plug)	$1{,}43 \angle 21{,}2^\circ$	$0{,}87 \angle 39{,}4^\circ$

* Actual estimated plug mass = 43 g

790190

Approx. value of rotating mass = 5 kg

Table 1. Values of Synchronous Vibration Component in μm (RMS) at 4800 RPM

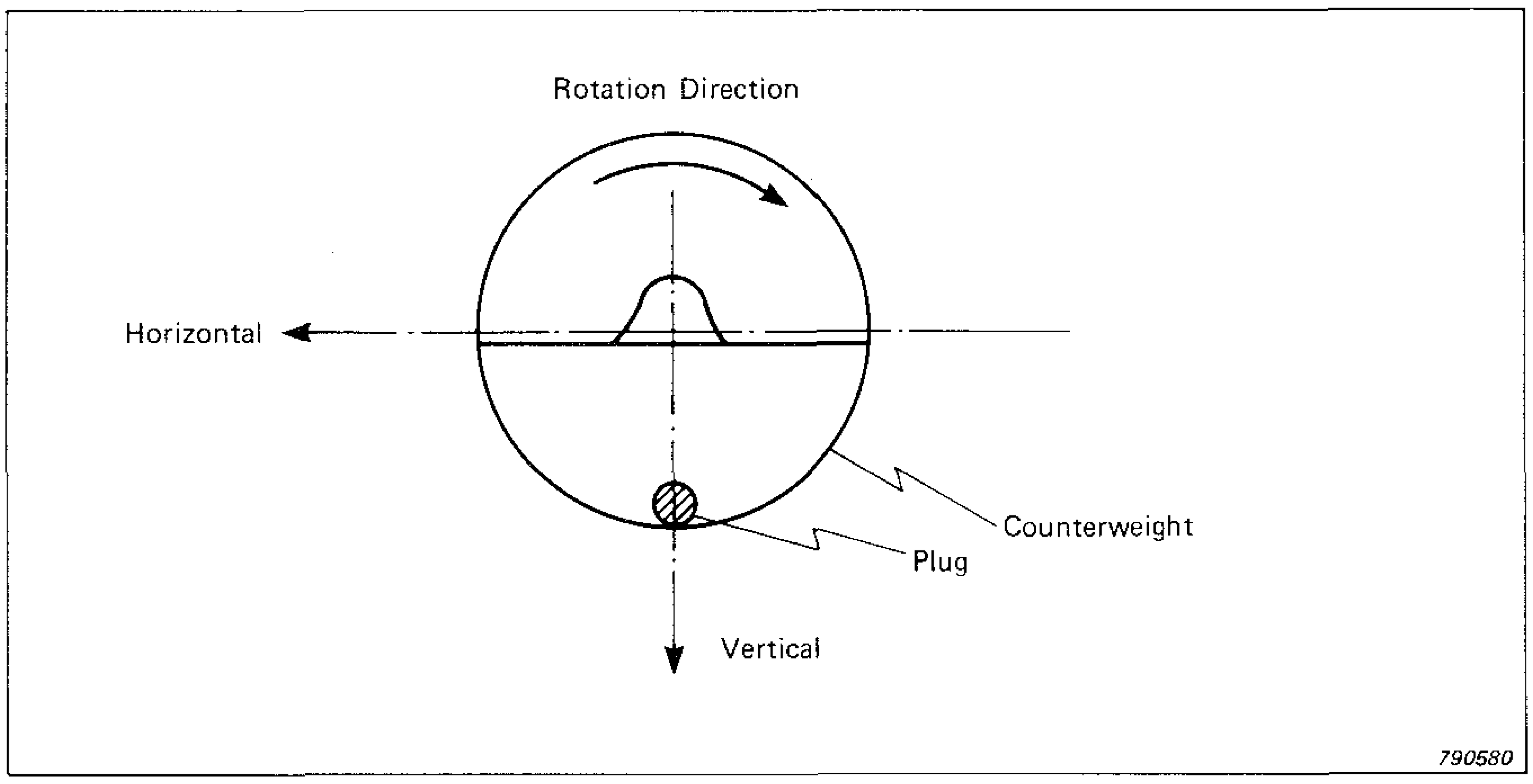

Fig.5. Separation of correction components

		Counterweight 1 (Top)	Counterweight 2 (Bottom)
4800 RPM	Vertical	1,33	0,67
	Horizontal	0,52	0,55
2800 RPM	Vertical	–0,11*	–0,97
	Horizontal	2,12	2,08

* —negative value is mass removal.

790191

Table 2. Values of Mass Correction Required as a Proportion of Plug Mass

For the horizontal component of correction, this is a pure couple correction at both speeds. This suggests that the motion is a function of the mounting clamp by which the engine was secured to the tank. The large compensating values of 2,1 indicated at 2800 RPM suggest that there is a resonance frequency in the mounting which is excited by lateral unbalance in the crank assembly. At the 4800 RPM test speed the average correction is 0,54, which indicates that the balance quality of the crankshaft itself might need to be improved at the production stage, or that there is asymmetry in the connecting-rod big-end assembly causing these effects. Alternatively, the final quality could be improved by increasing the torsional stiffness of the engine mounting assembly.

These measurements were made using a test tank and a stationary motor. In the future it is envisaged that similar work would be carried out in a boat powered by the motor under test, when the effects due to the tank size, such as the cavitation at the propellor, would be significantly reduced. The whole procedure could then facilitate this kind of study.

The use of such portable equipment is extremely useful for engine design and development, as realistic, full-scale tests can be made under normal operating conditions. Parameters such as balance factor, production balance quality, etc. can be investigated in terms of their effect on complete in-service performance at different speeds and loading conditions. The Type 9500 is also compatible with other instruments such as the Portable Level Recorder 2306, Magnetic Tape Recorder 7003, and Real-Time Analyzer 2031, allowing different types of analysis to be performed.

Jonathan Wort, Brüel & Kjær

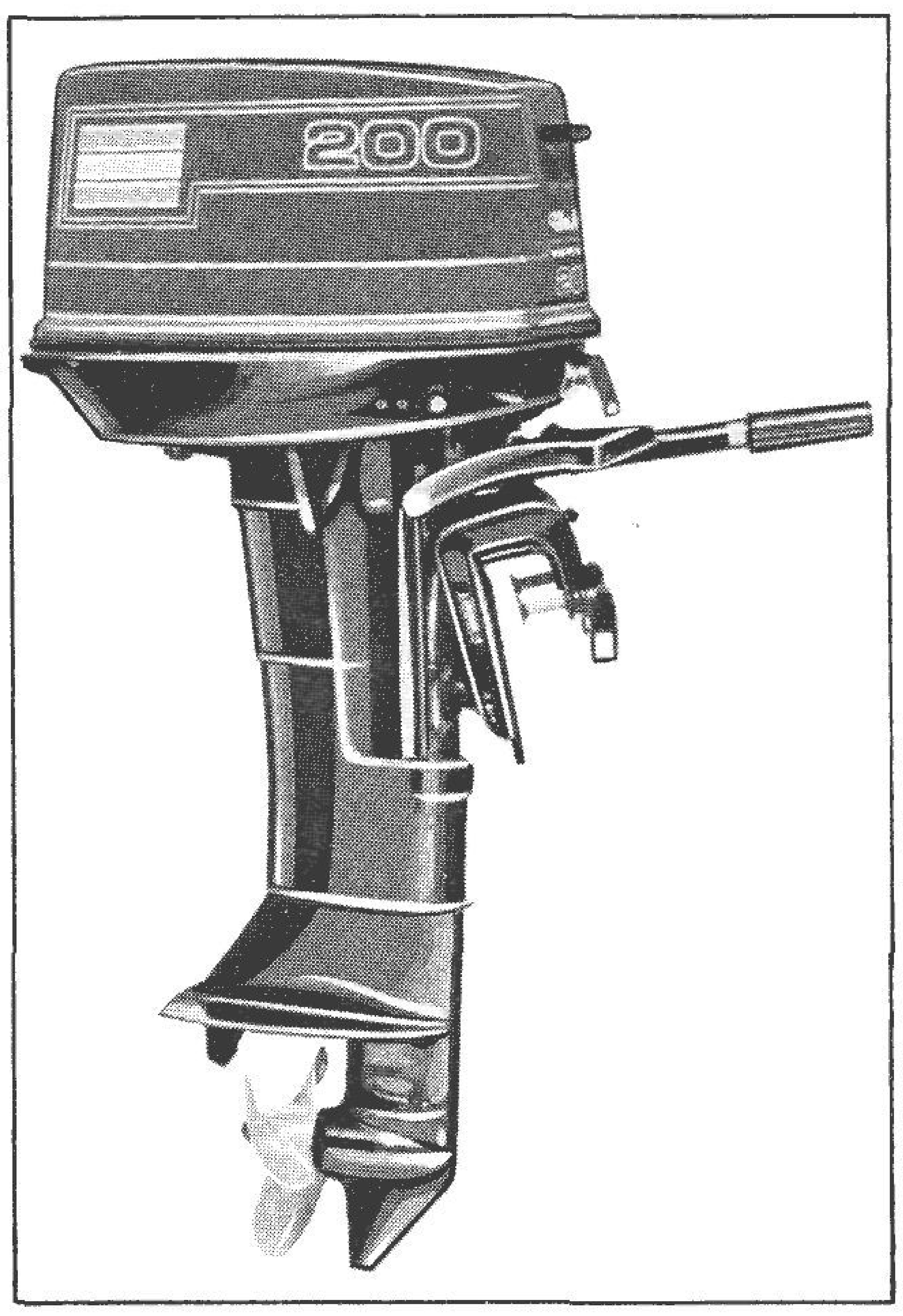

Fig.6. An outboard motor of a type similar to the one on which the measurements were made

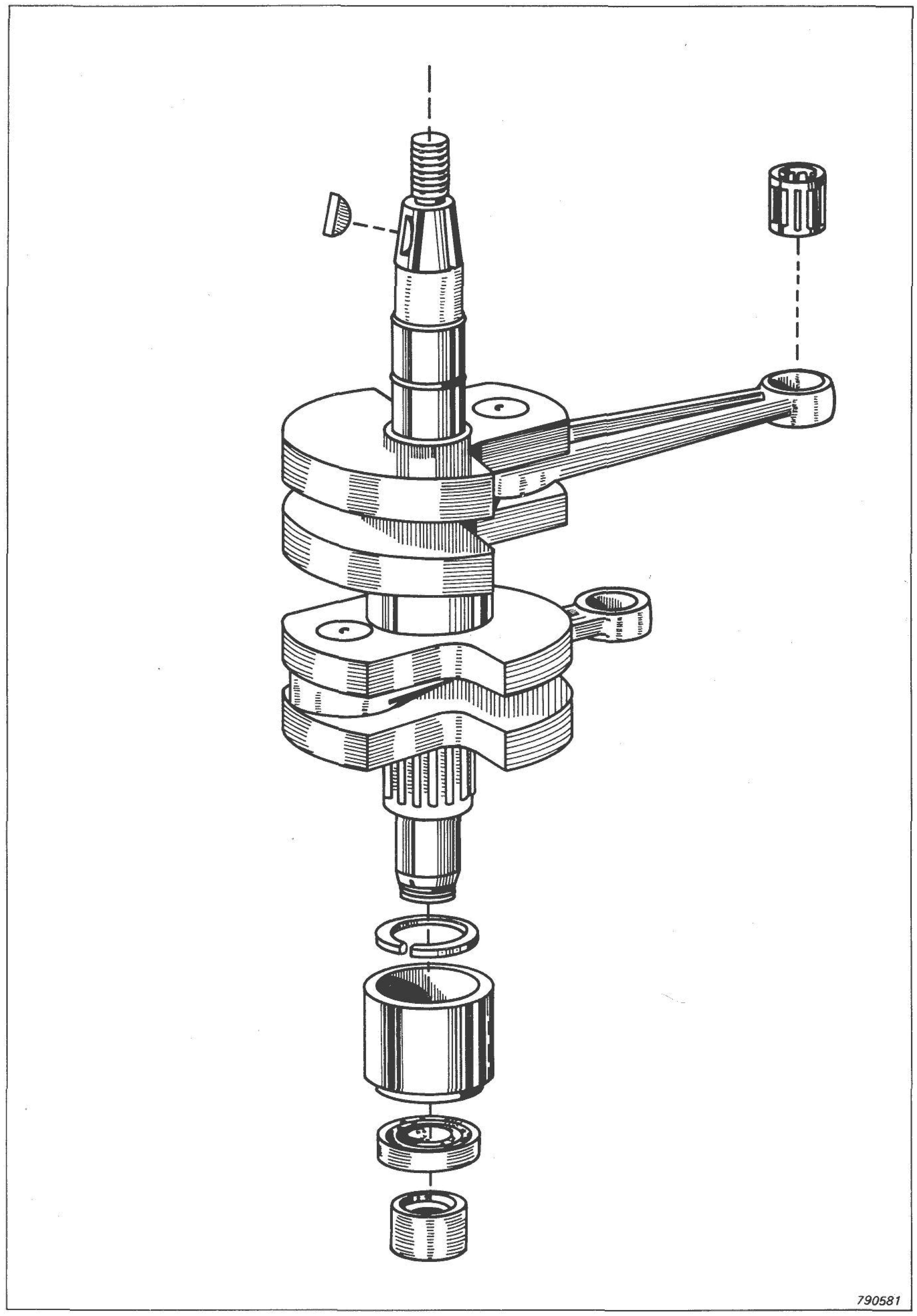

Fig.7. Exploded diagram showing a twin-cylinder crankshaft assembly

Acknowledgement:

Thanks are extended to Mr. Lars Andersson of Volvo Penta, Uppsala, Sweden for permission to describe this trial.

Bruel & Kjaer Instruments, Inc.

185 Forest Street, Marlborough, Massachusetts 01752 / (617) 481-7000 TWX 710/347-1187

Printed in Denmark by Nærum Offset