Y.100 MACHINERY CRUISING TURBINE CLUTCH

SEA EFFECTS

BY

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Many of the effects of ship motion on a ship's transmission system are well known. However, recent trials in H.M.S. *Whitby* revealed facts which had not hitherto been fully appreciated. The object of this article is to present a few of these facts, to indicate how they affect the Y.100 machinery automatic cruising turbine clutch and to show why a complete re-design of the clutch has been necessary.

Background

The concept of the Y.100 machinery requires a clutch to disengage the cruising turbine automatically on increasing above 29 per cent power (20 knots) and re-engage the cruising turbine automatically on reducing power through the same point. This entails a changeover point at approximately 8,000 r.p.m. of the cruising turbine (147 shaft r.p.m.). The original design to meet this requirement was a non-locking friction servo dog clutch.

The prototype machinery trials at Pametrada indicated that with minor modifications the clutch would be satisfactory, but the first sea trials soon revealed shortcomings in the design, leading at times to impact torques of damaging value to the gearing. The most disastrous blows occurred during manœuvring from astern to ahead when a violent shuttling action arose.

Investigation showed that the incorporation of a 'lock in' feature would overcome the majority of the defects including, of course, the manœuvring condition referred to above. Further, it was readily apparent that the data required to carry out a re-design with confidence was not complete.

The sea trials, which form the basis of this article, were specifically designed to obtain this data. The original design of non-locking clutch was used in conjunction with a safety fuze in the form of a torque limiting device.

In the light of data obtained from the first sea trials, the use of alternative designs of clutch (e.g. friction-hydraulic) was investigated. The studies showed that no design, other than the automatic mechanical type with a 'lock in' feature, was likely to be successful without long term development. Investigation also showed that it may be to advantage to fit the clutch at a point in the system other than the cruising train. However, this would entail a major re-design of the gearing.

The obvious line of attack, therefore, was the re-design of a mechanical type clutch for the cruising train. All effort is now being directed towards this end and it is intended to confine this article to similar lines. Two independent designs are now in hand and it is intended to manufacture two prototypes to each design for sea evaluation.

Mechanics

The problem is best illustrated by outlining the mechanics involved and indicating the significance of the various factors.

The transmission system may be shown in simple diagrammatic form as in FIG. 1.

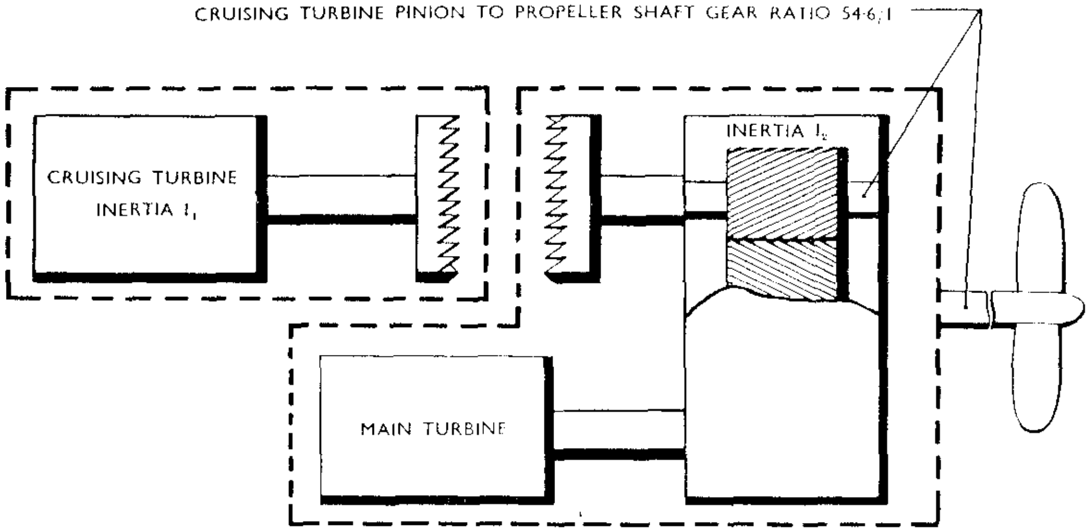


FIG. 1

The impact torque (T) on engagement, of this type of clutch is a function of :—

(i) The inertia of the cruising turbine and drive up to the clutch (I_1)

(ii) The inertia of the gearing, main turbine, shafting, propeller, etc., referred to the cruising pinion (I_2)

(iii) The relevant natural frequency of torsional oscillation of the whole transmission system (f) (in a ship's transmission system this will, in general, be the second node mode)

(iv) The differential speed of the engaging clutch members at the moment of impact (ω)

i.e. $T = F(I_1, I_2, f, \omega)$.

Considering these factors in turn :

(i) and (ii) I_1 and I_2 are, of course, almost entirely dictated by physical considerations other than clutch design, e.g. turbine thermo-dynamic and strength aspects, gearing design, etc.

(iii) The natural frequency of torsional oscillation, being a function of I_1, I_2 and torsional stiffness of the system (S) is largely determined by considerations such as shaft sizes required in the gearing. Investigations into varying the torsional stiffness of components in order to ease the clutch impact torque, without prejudice to other considerations, show that the frequency can be varied within certain limits.

(iv) The differential speed (ω) of the engaging clutch members at the moment of impact is a function of the geometry of the clutch and the differential acceleration of the engaging members during the period of engagement.

It is apparent that of the factors affecting T, I_1 and I_2 are not readily variable, f should be reduced to the minimum practicable, and ω must be such that T shall be of acceptable value.

Assuming that the clutch engagement operation commences at synchronous speed and is positive in action, then

$$\omega = F(\theta)$$

where θ is the angular rotation required to engage the clutch members measured from the fully out to the fully engaged positions.

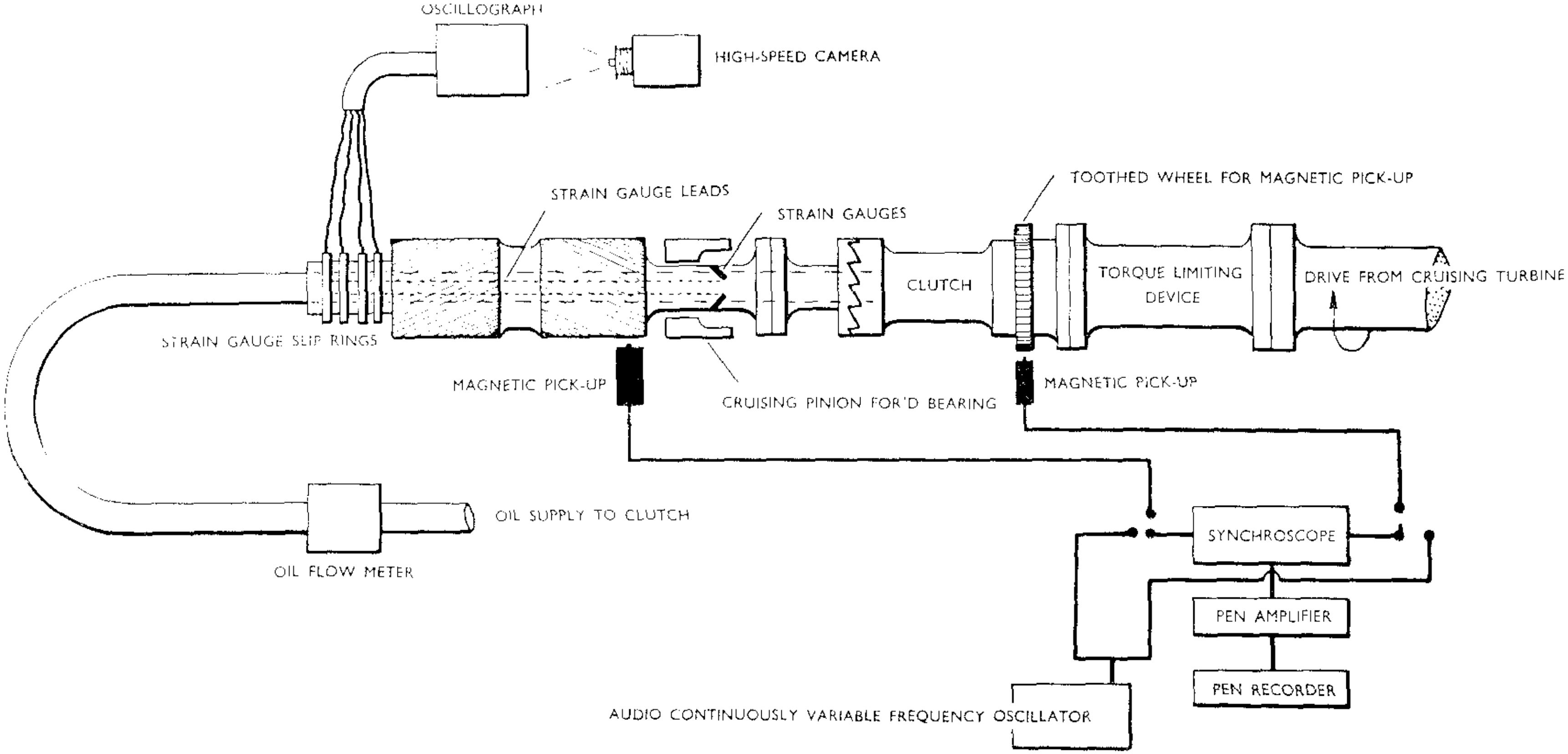


FIG. 2 SEA TRIALS TEST RIG AS FITTED TO STARBOARD ENGINE, H.M.S. 'WHITBY'

TABLE 1

Conditions during trial : Shaft r.p.m. 100
Sea swell 18 ft
Wind force 8, fine on starboard bow

Peak accelerations and decelerations of cruising pinion (r.p.m./sec.) recorded over a period of four minutes :

96	54	228	86	115	80	106

i.e. Maximum cruising pinion acceleration recorded = 228 r.p.m./sec. which gives a main shaft acceleration of 4·2 r.p.m./sec.

and $\dot{\omega}$ is the differential acceleration of the two engaging members obtaining during the engagement period.

In this particular application $\dot{\omega}$ is dependent on the following :—

(i) Acceleration of the cruising turbine. This is a function of I_1, cruising turbine windage characteristics and the control system, which has been so designed to limit acceleration to a low value.

(ii) Deceleration of the cruising pinion due to the reduction of main turbine power. This is a function of I_2, the inertia of the ship, ship resistance characteristics, etc.

(iii) Sea effects on the cruising pinion speed due to ship motion. This is, of course, dependent on the ship motion characteristics.

Sea Trials

The instrumentation, as shown in FIG. 2, was fitted to the starboard engine of the ship and allows measurement of :—

(*a*) Cruising turbine speed relative to known oscillator output

(*b*) Cruising pinion speed relative to known oscillator output

(*c*) Cruising turbine speed relative to cruising pinion speed

(*d*) Impact torque on clutch engagement.

Values of (*a*), (*b*) and (*c*) were obtained on a pen recorder trace from which differential speeds and accelerations could be determined. The impact torque was recorded by high speed photography of the strain gauge oscillograph trace.

Trials, on reduction of power through the changeover point, showed that :—

(i) By correct adjustment of the control system the cruising turbine acceleration could be controlled to a maximum of 50 r.p.m. per sec.

(ii) The maximum deceleration of the cruising pinion, due to reduction of main turbine power, would be 80 r.p.m. per sec.

Sea effects were then determined by steaming the ship on various courses relative to the sea and turning under different degrees of helm in various sea conditions. Examples of data obtained are given in FIGS. 3 and 4 and TABLE 1.

Full analysis shows that accelerations and decelerations are of the same order, are completely random in nature and, to cover all sea conditions likely to be encountered, a differential acceleration of ±400 r.p.m. per sec. must be allowed for on this account alone.

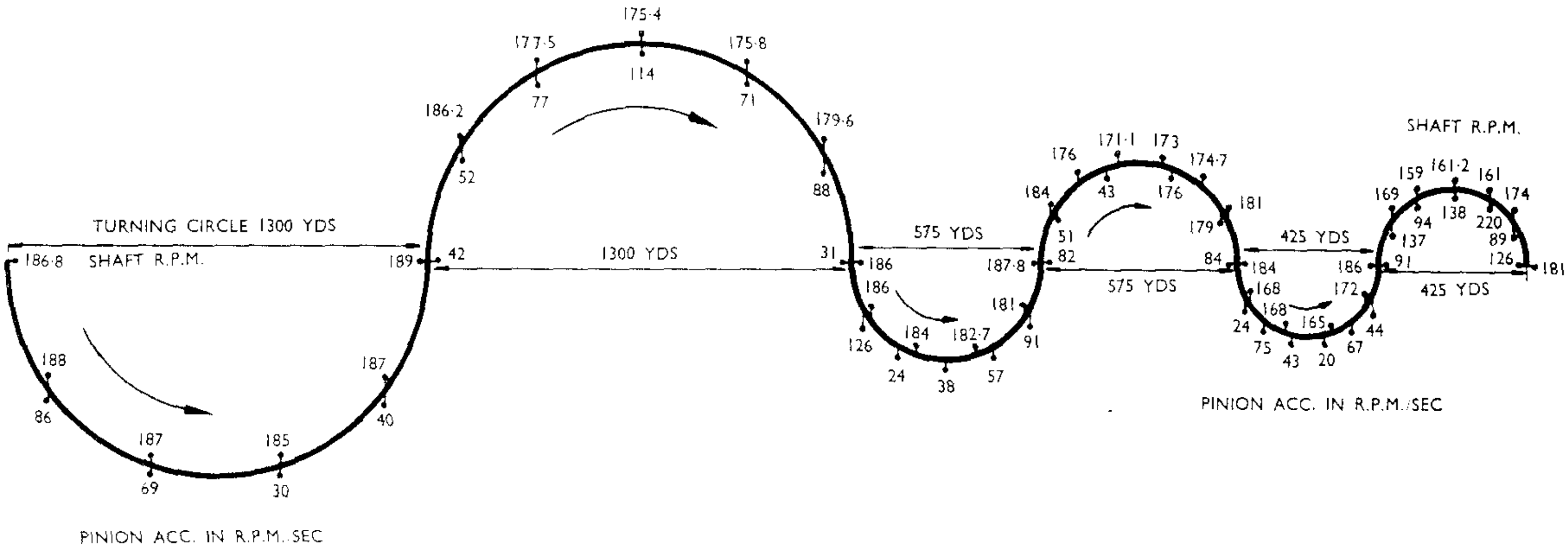


FIG. 3—TURNING TRIALS—VARIATION OF SHAFT R.P.M. AND CRUISING PINION ACCELERATION DURING TURNING

Ships speed 187 r.p.m. — 27 Knots
Weather conditions — Flat calm
Instrumentation carried out on Starboard shaft only

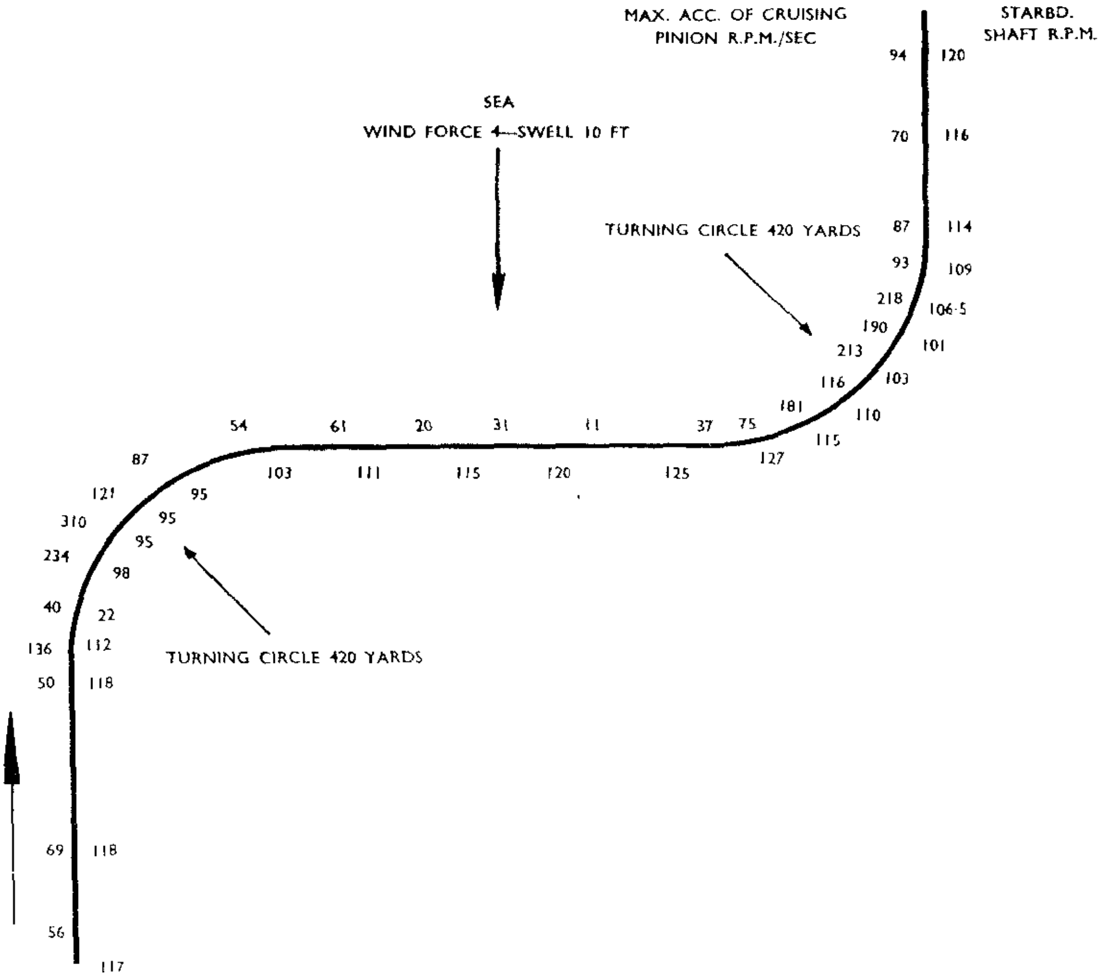


FIG. 4 VARIATION OF SHAFT R.P.M. AND CRUISING TURBINE ACCELERATION DURING MANOEUVRING

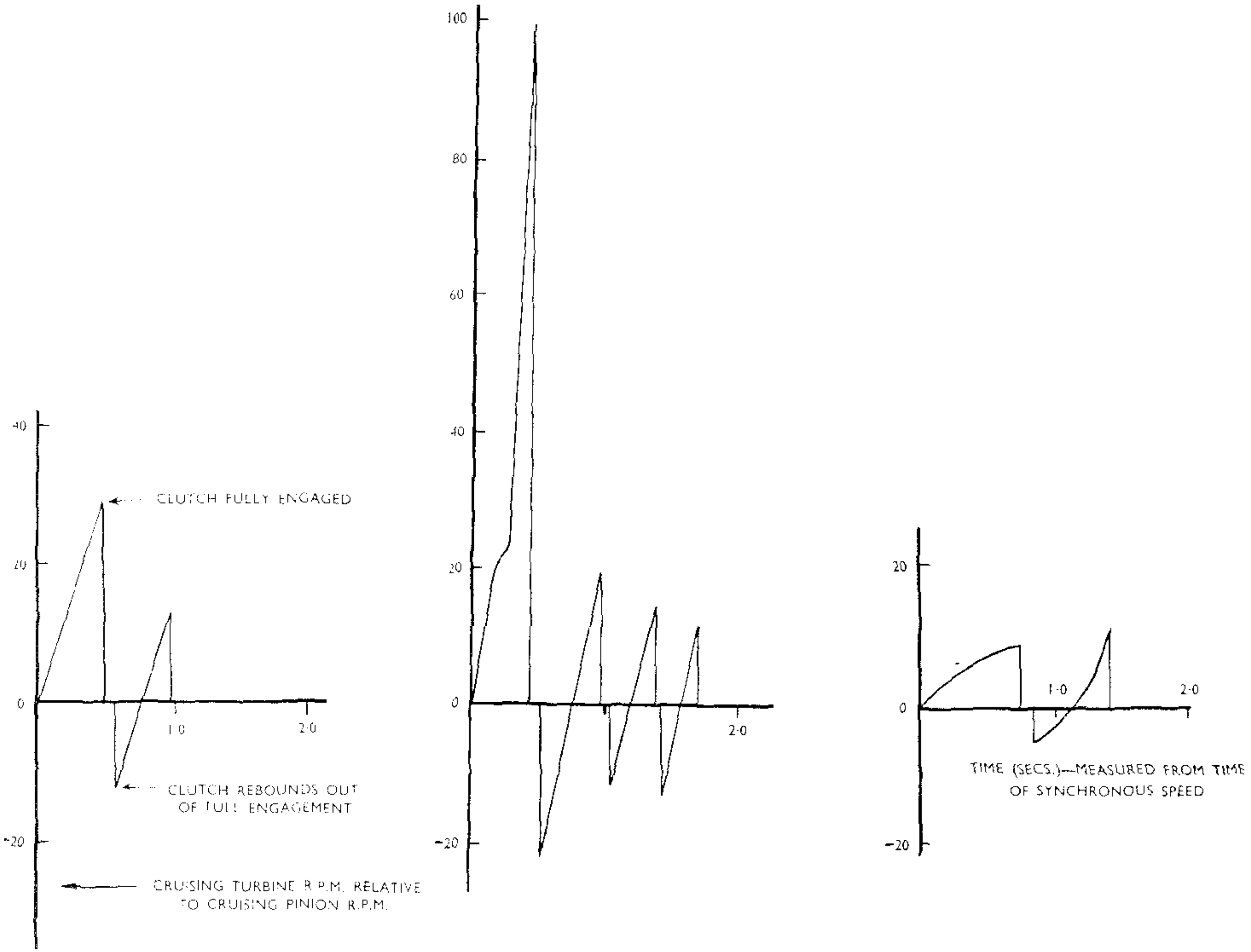
Manœuvre:
Ship steaming into head sea
30 degrees helm turn to Starboard—ship in beam sea for seven minutes
30 degrees helm turn to Port—ship steaming into head sea
Shaft r.p.m. on Telegraph—116
Throttle setting constant during manœuvre

Note:
Shaft r.p.m. are max. and min. values recorded. Cruising turbine acceleration are the max. values recorded at the points indicated.

$$\text{Gearing ratio} \; \frac{\text{Cruising Pinion}}{\text{Shaft}} \; \frac{54{\cdot}6}{1}$$

Clutch engagement trials were carried out under identical conditions from the operator aspect but under different weather conditions. On analysis, as one would expect, a damped oscillation, as the clutch rebounds out of engagement, is readily apparent, but note the significance of the sea effect in the resulting differential speed and hence impact torque. See FIGS. 5 (*a*), 5 (*b*) and 5 (*c*).

Thus the problem becomes not only that of designing for an operator-controlled acceleration of 130 r.p.m. per sec., but also a further random, unpredictable and uncontrollable acceleration of $\pm$ 400 r.p.m. per sec. It is



(*a*) Sea—Flat Calm

(*b*) Sea—18 ft Swell
Wind—Force 5

(*c*) Sea—18 ft Swell
Wind—Force 5

FIG. 5—ANALYSIS OF CLUTCH ENGAGEMENT TRIALS

Note:

(i) The above trials were carried out under identical conditions from the operator's aspect but observe the significance of the sea effect on the cruising pinion ((b) additive and (c) subtractive)—hence differential speed at engagement

(ii) The resulting impact torque—(lb ft)—is given by differential r.p.m. × 175

(iii) Maximum design cruising turbine driving torque is 4,600 lb ft the magnitude of the latter figure compared with the former that necessitates a complete re-design.

It follows that the success of the clutch will largely depend on achieving engagement within as small an angular rotation as possible. This is no mean problem when one views the physical aspect of overall space available (cylinder $12\frac{1}{2}$ in. long x 11 in. diameter) related to the power to be transmitted (2,300 h.p. at 7,000 r.p.m. and torques up to 4,600 lb ft) and the differential accelerations concerned. Incorporation of the ' lock in ' feature is also a significant problem in itself. Further, the magnitude of the sea effect compared to the operator controlled acceleration precludes the use of an anticipatory or ' bias in ' device.

The fact that the worst conditions may arise during a ' rugged middle watch ' is also fully appreciated and the control system has been designed accordingly.

Post Script

Successful sea evaluation of one design has now been carried out. The two clutches to this design will be left in service for about six months before finally accepting the design as proven.

Sea evaluation trials of the second design are programmed for April/May, 1959.