

## CRUISING TURBINES OF THE 'Y100' NAVAL PROPULSION MACHINERY

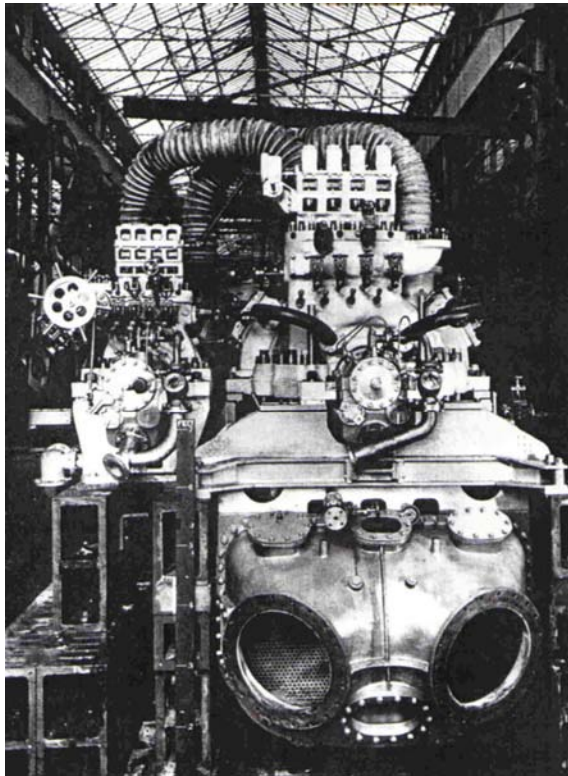
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### Introduction

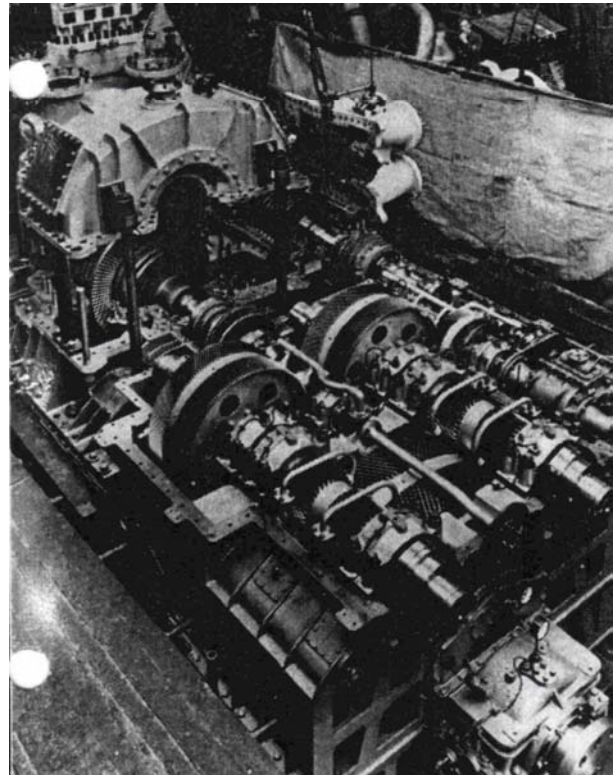
The Yarrow Admiralty Research Department (YARD) / English Electric 'Y100' marine steam turbine plant is considered to be an all time classic design: several hundred machinery installations of this type were built for navies throughout the world, and the design continued to be manufactured more than 40-years after its original inception.

While undeniably successful, an important part of the originally perceived benefits of this machinery was lost at an early stage when cruising turbines were deleted from the design.

The commonly cited reason for the abandonment of the cruising turbine concept is the failure of the automatic clutch contained within the cruising turbine drive train to operate satisfactorily. This failure has served to obscure further deficiencies that were inherent in the original machinery design.



*Starboard 'Y100' Machinery Set*



*Turbine Casings & Gearcase Covers open*

### The Requirement

In the late 1940's, the Admiralty became interested in the possibilities of an Anti-Submarine escort vessel of some 2,000 tons displacement, to be installed with main machinery of some 30,000 s.h.p. on two shafts. This new requirement for a fast Anti-Submarine frigate was a direct result of the Korean War. A speed sufficient to deal with modern submarines was required, together with a very long endurance at speeds in the region of 15 knots when providing merchant convoy escort.

This latter requirement came about from an analysis of World War II destroyer operations: this revealed that they spent at least 80% of their life at cruising speed or less, and, consequent upon the characteristics of a destroyer's power / speed curve, this meant that 80% of a ship's life was spent at less than 20% of the installed power. Given this knowledge it became obvious that the use of full power as the machinery design point could no longer be justified; the overriding requirement was to reduce fuel consumption and increase endurance at the designated economical speed.

The displacement of the new escort vessel had to be restricted to allow the necessary manoeuvrability for anti-submarine work; this, in turn, severely limited the permitted weight of machinery, plus fuel, to give the specified endurance. The Director of Naval Construction had stated that such a ship could only be achieved if the combined weight of propulsion machinery and fuel did not appreciably exceed 660 tons. By comparison with wartime practice this requirement demanded a very substantial advance in machinery weight, space, and efficiency standards. Based on the *Dido*-Class cruiser machinery of 15,000 s.h.p. per shaft, the reduction in weight would have to be approximately 33%, with a corresponding reduction in space, if the requirements were to be fulfilled. This represented a 25% reduction in the specific weight of the most advanced propulsion machinery then operated by the Royal Navy, that of the *Daring*-Class, as well as an improved fuel consumption at the lower speeds.

A specification for the new plant, designated 'Y100', for 15,000shp (11,200kW) on each of two shafts was issued early in 1949. A few months later, the Royal Canadian Navy decided to fit the same design of machinery in their new class of escort vessels; all but the first set would be manufactured in Canada and the first set, built in the UK, had to be shipped up the St Lawrence Seaway to Montreal before the river froze in the winter of 1951.

Three firms competed for the turbine design. The tender from the English Electric Company was considered by the Admiralty to be the most suitable, and a contract for the design and manufacture of one set of prototype machinery was placed. This prototype machinery set was to undergo full-scale shore trials at the PAMETRADA test house, Wallsend-on-Tyne.

### Specific Requirements

- The full power of 15,000 h.p. per shaft to be obtained when supplied with steam to turbine nozzles at 450lb. per sq. in. gauge and 825 deg. F., the vacuum at exhaust being 23-in. Hg. under tropical conditions with a cooling water temperature of 85 deg. F.
- The ahead turbines were to attain a high efficiency at all powers between 5% and 100%; a good performance over the range of 5% to 20% being particularly important.
- The astern turbines were to produce 5,000 s.h.p. under both tropical and temperate conditions.
- Machinery weight and space to be kept to a minimum.
- Simplicity of manufacture to allow for mass-production.
- Suitable for manufacture by licensees.
- Avoidance of relatively scarce materials.
- The design to be one which could be operated by wartime conscripts with comparatively little training.

### Advanced Machinery

These onerous requirements influenced all aspects of the 'Y100' steam plant design and required the introduction of several features then novel to the Royal Navy.

- Increased rates of boiler furnace forcing with fuel oil being fired in high draught loss air registers.
- The employment of flue gas by-passing for superheat temperature control.
- The condenser was built integral with the main turbine casing bottom half.
- Slow-running – 220 r.p.m. versus the previous norm of 350 r.p.m. – high efficiency 12-foot diameter propellers.
- Auxiliary systems were simplified and where possible automated. Automatic control of turbine shaft gland sealing steam pressure and turbine lubricating oil temperature was included.



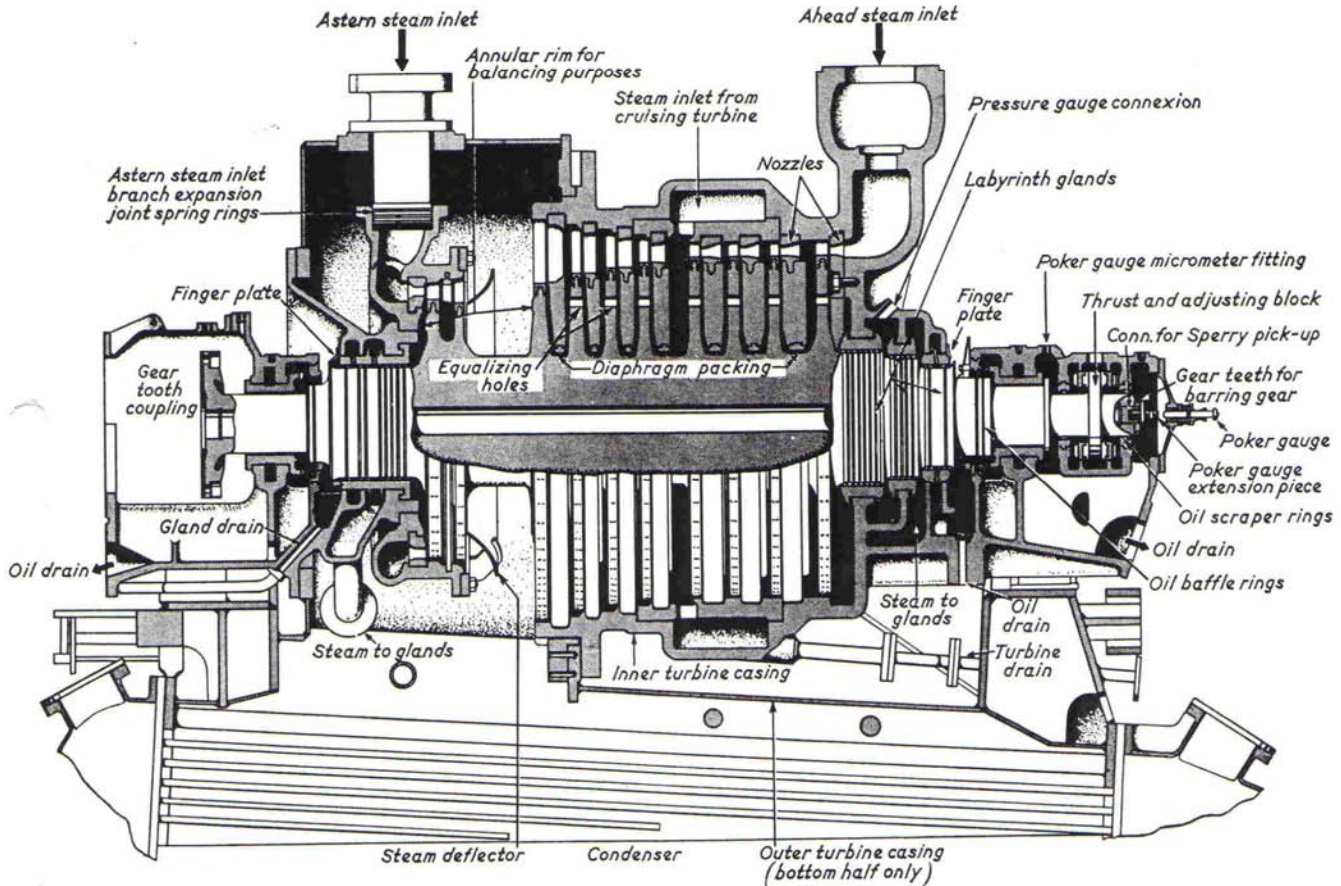
*HMS Plymouth Port Main Turbine  
(Hull Stabiliser Hydraulic Pump in lieu of Cruising Turbine)*



## Factors Influencing the Selection of Cruising Turbines

The design chosen for the main turbine comprised of a light-weight, eight-stage impulse-bladed ahead unit, incorporating a two-row Curtis Wheel astern turbine within a common casing. The requirement of high efficiency at low powers without prejudicing the performance at full power also made it essential to provide some form of arrangement incorporating a cruising turbine.

Preliminary investigation work made it obvious that a de-clutchable cruising turbine possessed considerable advantage over one that is permanently coupled. The former could be de-clutched at some appropriate point along the power curve when its efficiency fell below that of the main turbine, thus at higher powers all rotational losses that would otherwise have been increased were eliminated.



Y100 Main Turbine cross-section

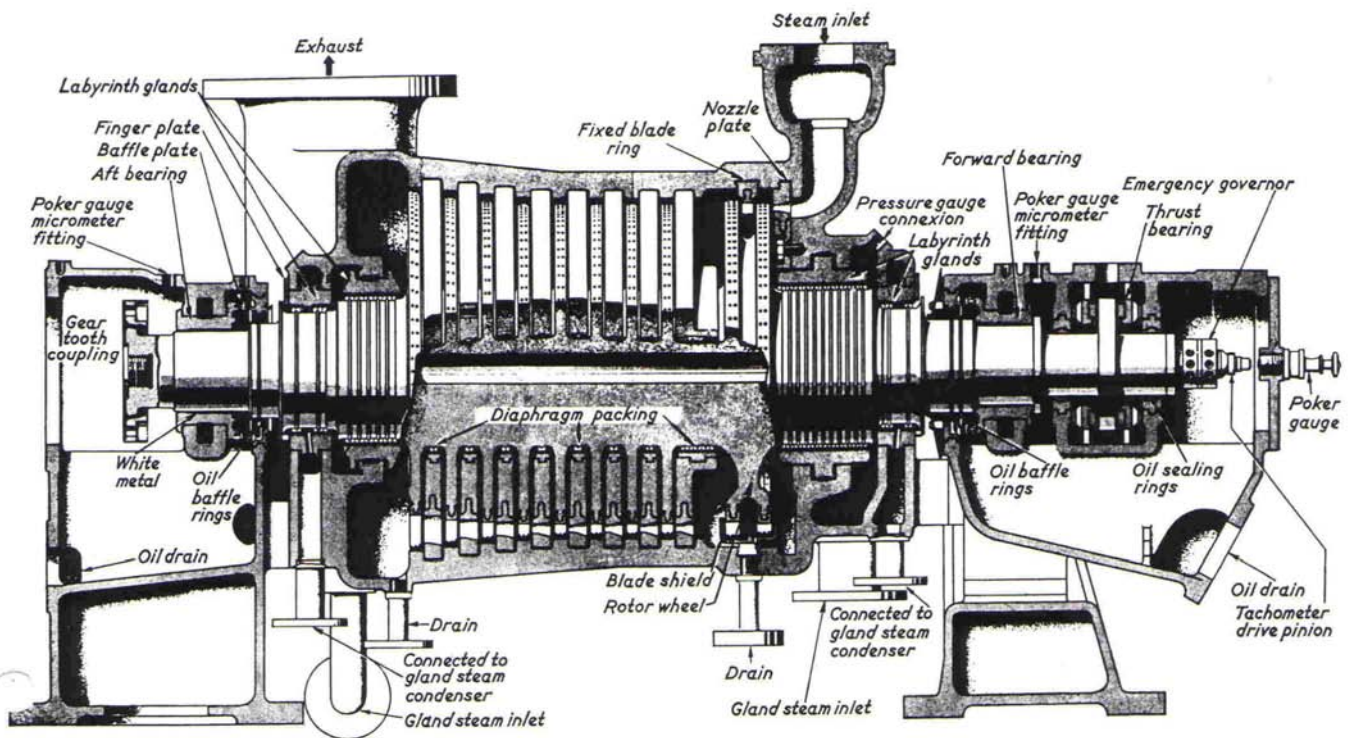
A de-clutchable cruising turbine also affected an appreciable saving in weight and space due to the design being based on a maximum speed corresponding to some pre-determined part load, whereas if permanently coupled the turbine would have had to be designed to withstand the stresses encountered at the higher speed corresponding to full power.

The added complication entailed by the incorporation of the necessary automatic mechanical clutch was considered to be more than offset by improved performance combined with reduced weight and space.

## Design of the Cruising Turbine

The cruising turbine was of all-impulse design, comprising a double-row Curtis Wheel followed by eight single impulse stages. The steam chest was cast as an integral part of the top half casing and contained an emergency stop valve and four mechanically operated nozzle control valves, each of which supplied steam to individual nozzle groups, all four groups being in the casing top half. The exhaust steam was led through two cross-over pipes into the fifth stage of the main turbine via a bypass belt.

The cruising turbine was engaged for all powers up to approximately 33%; all load being transferred to the main turbine at powers above that point. The change of load from one turbine to the other was carried out by means of the manually operated nozzle control valve mechanism in conjunction with the clutch, which was to have automatically engaged and disengaged the cruising turbine at that point in the power range where the efficiency of the cruising turbine fell below that of the bypass stages in the main turbine.



*Y100 Cruising Turbine cross-section*

### Turbine Valve Gear

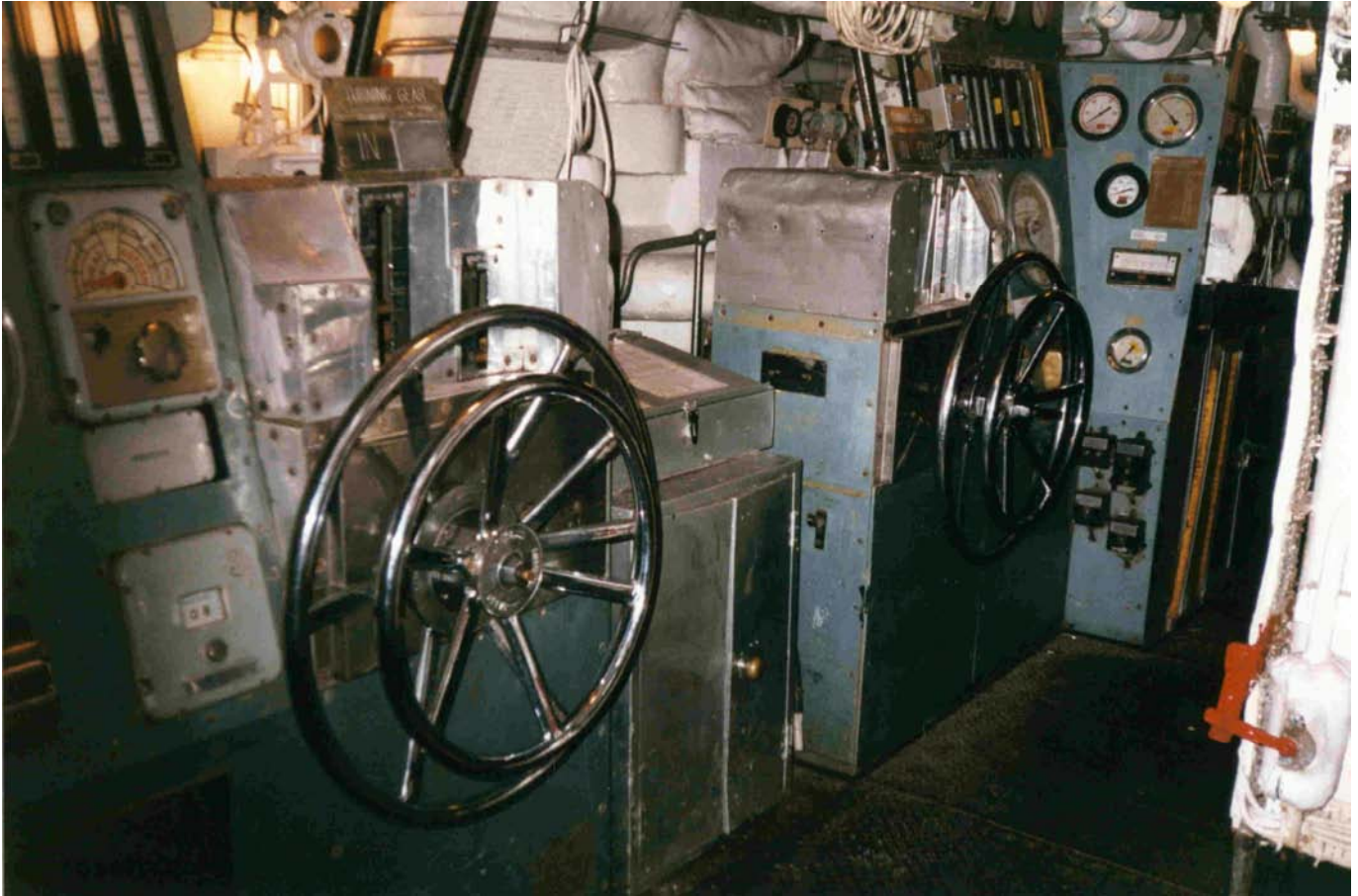
The arrangement of partial admission belts and individual nozzle control valves minimised losses due to steam throttling over partially open valves. Main and cruising turbines were both controlled throughout the entire power range by means of a single Ahead hand wheel that sequentially operated the two groups of four nozzle valves associated with each turbine via cam lift mechanisms and suitable shafting, flexible couplings, and worm gearing.



*HMS Plymouth Starboard Turbine Nozzle Valve Gear*



Shaft speed was increased by turning the Ahead hand wheel in the anti-clockwise opening direction, which opened the four cruising turbine nozzle control valves sequentially until, at approximately 25% power, all four were fully open. Until this point was reached, the steam from the cruising turbine exhausted into the bypass belt immediately upstream of the fifth stage of the main turbine, so bypassing the first four stages. Therefore all power was being produced jointly by the cruising turbine and the last four stages of the main turbine.



*HMS Plymouth Main Engine Starting Platform*

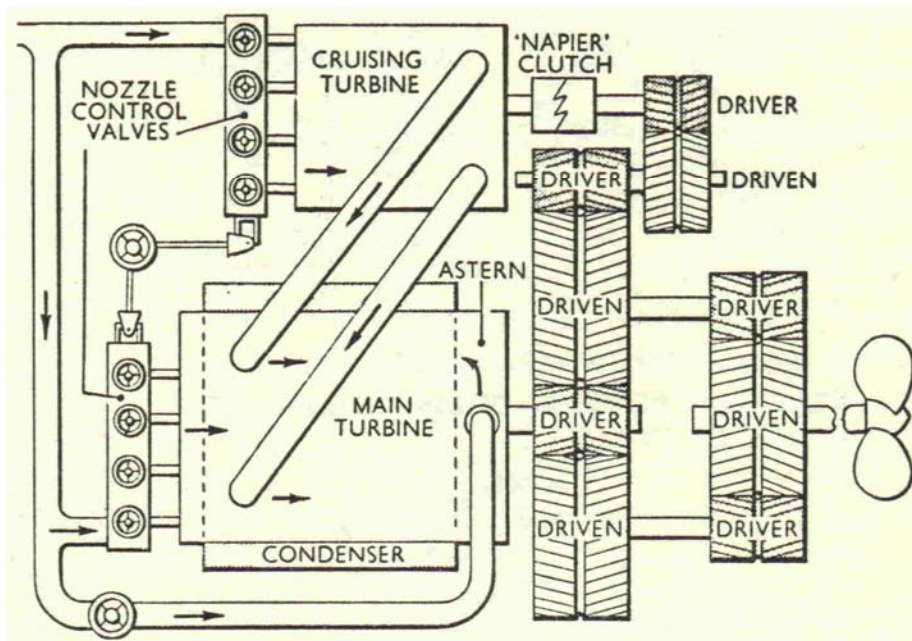
Further opening of the Ahead hand wheel rapidly closed the cruising turbine control valves and simultaneously opened the first main turbine control valve, followed sequentially by main turbine nozzles Nos. 2, 3 & 4. Thus, at all powers above approximately 33%, all steam was admitted to the first-stage nozzles of the main turbine, except for a small quantity that continued to be admitted through a pilot valve incorporated integrally with cruising nozzle No. 1. The purpose of this pilot valve was to maintain a "rolling steam" supply to the cruising turbine sufficient to turn it at around 400 r.p.m., so preventing rotor hogging through imbalanced heating when the cruising turbine was disengaged.

In order to produce a smooth change when transferring power from the cruising turbine to the main turbine, between approximately 25% and 33% power, steam was admitted simultaneously to the nozzles of both turbines. The same overlap conditions applied when decreasing power to cruising levels.

### **Napier Cruising Turbine Clutch**

While the main turbine was geared to the propeller shaft by double reduction locked train gearing, the cruising turbine train comprised of an additional reduction gear meshing with the outboard first reduction wheel of the main locked train reduction gear set. An automatic non-locking friction servo dog clutch, designed and manufactured by D. Napier and Son, Ltd., was interposed in this cruising train, intended to engage, or disengage, according to the relative speeds and power outputs of the cruising and main turbine shafts.

When increasing power the clutch was to have disengaged as soon as the steam quantity to the cruising turbine was insufficient to maintain it running at the synchronous speed of the main turbine. When reducing power, conversely, the clutch was to have engaged when the steam flow to the cruising turbine was sufficient to take it up to the synchronous speed of the main turbine. Thereafter, it was to have remained engaged for so long as there was a power output from the cruising turbine.



*Y100 Machinery Block Plan*

The intended operation point of the clutch corresponded to a cruising turbine speed of 8,000 rpm (147 shaft revolutions), or some 29% of full power (approximately 20 knots). In practice, the actual speed in revolutions at which the clutch engaged or disengaged was dependant upon various factors. These factors were the nozzle valve and valve operating cam profiles and the relative efficiency of the cruising turbine. It was nevertheless intended that the characteristics of the clutch would at all times ensure an appropriate speed of engagement / disengagement consistent with safe stress levels and adequate factors of safety for all rotating parts.

Although the original intention of the design was that the clutch should be fully automatic in operation, in the event of any damage occurring to the cruising turbine, the ship was to have continued to be operationally available with power being obtained from the main turbine alone. To enable this mode of operation, a device was incorporated in the Napier clutch whereby it could be withdrawn manually and held in the disengaged position.

Manual disengagement of the clutch was accomplished by winding a hand wheel, the initial movement of which tripped the cruising turbine emergency stop valve. This first action was necessary to prevent the over speeding which would have otherwise occurred when load was removed from the cruising turbine following clutch disengagement.

It is fortunate that the facility to manually disengage the cruising turbine clutch existed for, during initial sea trials, shortcomings in the design of the automatic clutch quickly became apparent. Under certain ship's manoeuvres, when adopting a zigzagging course at some powers, or when manoeuvring from ahead to astern, a violent "shuttling" action of the clutch arose, with its repeatedly cycling in and out of engagement. In this latter case, as the turbines ran in the reverse direction, the clutch would, if permitted, engage automatically and therefore the cruising turbine would overheat due to blade windage and disc friction during prolonged high speed astern operation.

The prevention of these unintended clutch operations necessitated the added complication of the clutch having to be placed in the manually withdrawn position when the ship was undergoing operations involving rapid manoeuvring, or when steaming astern. This manual intervention not only undermined the principle of single-point sequential control of the entire power range from the engine starting platform, but moreover, if the clutch was not re-engaged when returning to steady cruising conditions, then the considerable expense of increased steam consumption was incurred due to the decrease in overall efficiency of the main engines and heavy throttling losses that occurred below 27% power.

To achieve the desired duration at cruising speeds, it was essential that the cruising turbine was properly re-engaged. This could be achieved by winding the clutch hand wheel to the "engaged" or "automatic" position at any time except when running astern, following which the emergency stop valve could be reset and steam admitted to the cruising turbine via the nozzle control valves.



## Entry into Service

The 'Y100' machinery first put to sea in Royal Navy service in the 'Y101' variant 15,000 s.h.p. single shaft Type-14 frigate HMS Hardy in early 1956, slightly preceded by the Royal Canadian Navy's ship HMCS St Laurent, commissioned in October 1955. The first RN twin shaft Type-12 *Whitby*-Class vessel HMS Torquay was completed in May 1956.



*HMS Torquay (as built)*

At first the new machinery design caused serious operator problems. The compactness of layout, higher rotational speeds, higher temperatures and pressures, and smaller system capacities all meant that the margins for error were smaller than those in the older and more conservatively rated machinery on which most operators had been trained.

The performance of the cruising turbine clutch was of particular concern and the failures that occurred established some clear principles: -

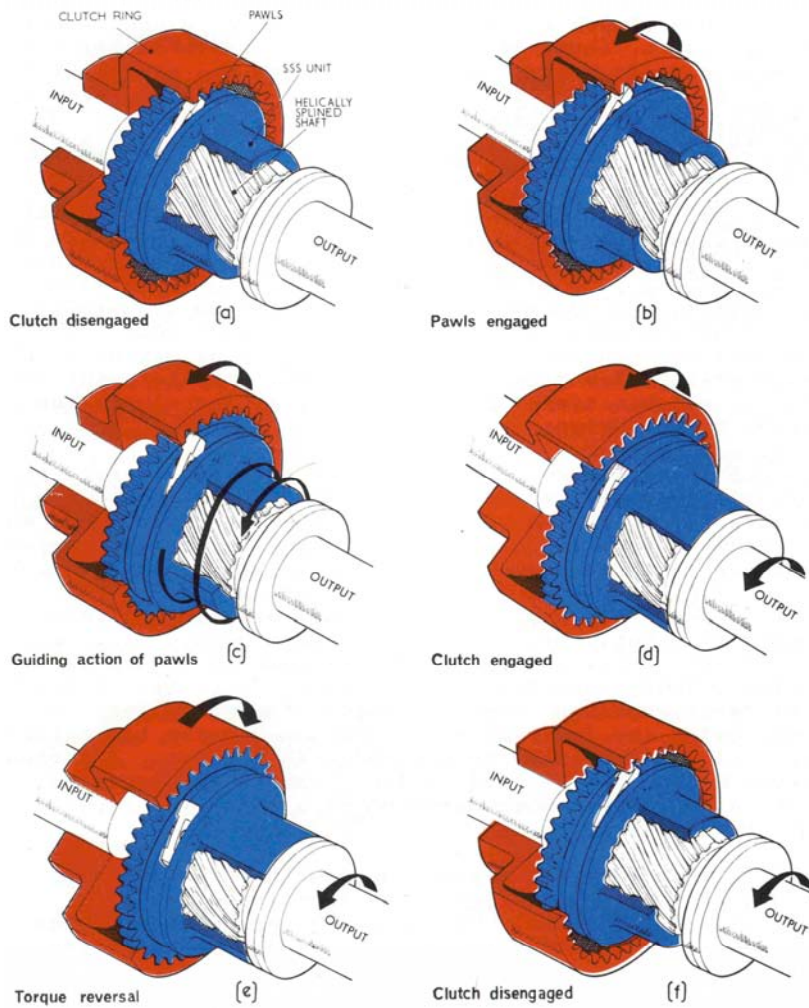
- A clutch relying solely on friction was unsuited for the powers and high rotational speeds associated with naval steam turbine plant.
- The effects of main shaft speed deceleration in a seaway at the moment of clutch engagement can be significantly magnified by the gear trains and can result in excessive differential accelerations of the opposing clutch members when the clutch is mounted in a high speed gear line.
- That without a suitable "locking-in" arrangement a marine clutch for main engines, which have large inertias, can "shuttle" during the moment of engagement, with consequent damage to the rotating mating elements.

These facts were of serious consequence to the designers of marine machinery: not only did they impact on the current 'Y100' steam machinery, but a correctly functioning clutch was a vital component of the proposed Combined Steam and Gas Turbine (COSAG) plants then under consideration for the Royal Navy's new Guided Missile Destroyers and General Purpose Frigates.

### **The Synchro-Self-Shifting ('SSS') Clutch**

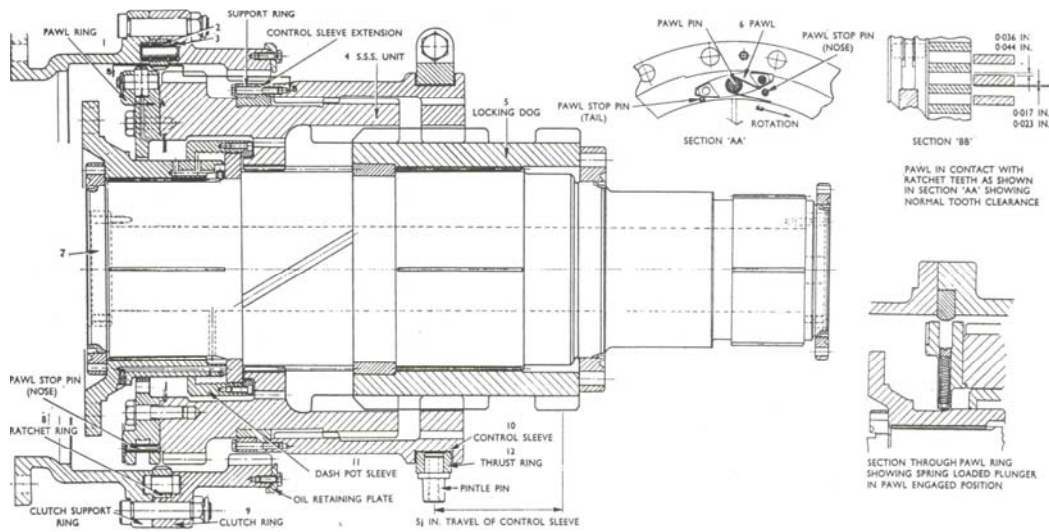
A new design of "Synchro-Self-Shifting" ('SSS') type clutch, with positive pawl actuating means, was therefore developed, and tested in a marine power train at a Shore Trials installation of the Barton Works of Associated Electrical Industries Limited. This clutch was fully automatic: it shifted into engagement on synchronism and locked itself in the engaged position, and; under all conditions of torque reversal, the SSS clutch would disengage.

The clutch comprised of three main sub-assemblies: an input assembly; a helically splined sliding sleeve assembly, and; an output assembly. Spring-loaded pawls acting on a ratchet sense the relative speeds of the input and output shafts. When the input shaft is about to overtake the other, the pawls engage with the ratchet and reactive torque is applied to the helically splined sleeve. Being free to slide, the sleeve assembly moves axially and brings the load-carrying clutch teeth into engagement. During this movement, the only load taken by the pawl is that required to shift the lightweight sliding component along the helical splines. The position of the pawls and the ratchet teeth ensure that the clutch teeth pass between each other exactly, without making contact until almost full engagement is reached. At the moment of clutch engagement, the pawls leave the ratchet teeth and cease to carry any further load. The SSS clutch can therefore operate continuously engaged or continuously overrunning at maximum speed without wear occurring.



*Basic 'SSS' Clutch – Principle of Operation*

Immediately on commencement of shore trials, clutch failure was experienced; again as a result of clutch engagement with a negative differential speed between the driving and driven members. This confirmed the sea experience previously gained on the *Whitby*-Class Napier cruising turbine clutch: it was essential for a clutch to be “locked-in” immediately after engagement, and, conversely; a clutch had to be “locked-out” on disengagement. Both of these things were essential towards ensuring that the clutch would not “shuttle” under any conditions.



*Fully-Developed 'SSS' Clutch with Control Sleeve*

Investigation showed that the basic clutch was not foolproof against all possible conditions of operation and a further control sleeve was introduced into the clutch assembly to fulfil the function of positively locking the clutch pawls in and out of engagement.



Since this control sleeve could be remotely actuated by hydraulic means, clutch lock-out could be readily incorporated in a scheme of sequential load control and the aim of a fully automatic clutch had been achieved.

Two 'Y100' series cruising turbine installations, those of HMS Scarborough (Type-12) and HMS Keppel (Type-14), were fitted with SSS clutches and these gave satisfactory service at sea.

The perfected SSS main synchronizing clutches were fitted as standard in both the contemporary 'Tribal'-Class general purpose frigates ('Y111A' machinery), and the 'County'-Class guided missile destroyers ('Y102A' plants) and they had soon completed several million hours of operation, requiring very little maintenance. Why then was the SSS clutch not included in the all-steam machineries of the subsequent Royal Navy *Leander*-Class and RCN destroyers?

### **Abandonment of 'Y100' Cruising Turbine Concept**

Aside from the problems associated with the 'Y100' cruise train clutch, which had been adequately resolved with the development of the SSS-type clutch by 1959, there was another more fundamental reason for the abandonment of the 'Y100' cruising turbine. This had its root in the design chosen for the cruising turbine itself, resulting in a disappointingly high steam consumption at low cruise powers.

While the overall cycle efficiency of a steam turbine plant is dictated by the inlet and exhaust steam conditions, the designer of any turbine must strive to ensure that the turbine efficiency is optimised to minimise internal losses between these design points. Important sources of these losses are: -

- Leakage of steam between rotating and stationary elements – these are minimised by ensuring that inter-stage leakage paths are small compared to the flow area through the blades and nozzles.
- Partial Admission Losses – the flow area at any point in the turbine is proportioned to match the specific volume and velocity of the steam at that point. If the full potential 360 degree inlet annulus is not used, then this is termed "partial admission" and losses occur as a result of the high velocity steam at the ends of the active arc entraining with "dead" steam.
- Passage Losses – these losses are proportional to the wetted surface of the steam flow passage in the blades and nozzles. For any given rotor diameter, the areas of the roof and platform of the steam passage essentially remains constant independently of blade height. If rotor diameter is increased, then the blades must be made shorter to retain the same cross-sectional flow area and these passage losses proportionally increase as a result of this reduction in height. It follows that the aim is to optimise rotor diameter and blade height.

From the above it will be realised that the primary aim is to achieve a well proportioned steam passage through the machine with the minimum necessary use of partial admission.

Furthermore, for an impulse turbine design, the maximum stage efficiency is attained when the mean blade velocity is around one-half of the steam velocity.

These factors in conjunction dictate that, if the diameter is reduced in order to obtain an acceptable blade height, then the shaft rotational speed must be increased in order to maintain the peripheral blade speed relative to that of the steam. This can only be done within those limitations imposed by rotor stress, which varies as a result of the square of the speed, and the ability to integrate a higher speed input shaft into the remainder of the gear train.

A mean blade diameter of 22-inches was chosen for the 'Y100' cruise turbine, with a speed of 8,510 r.p.m. when delivering its maximum output of 3,200 s.h.p. With a blade height of 0.81-inch having been selected for the first stage, consideration of the required steam flow area dictated the use of only 10 inlet nozzles out of a potential full admission arc of 80. This corresponded to 12.5% admission versus the optimum condition of 100% admission. As the steam expanded through the subsequent stages, full admission was not achieved until stage-5, where the blade height was 0.79-inch. The design chosen delivered a disappointingly high steam consumption of 8.24lb./s.h.p.-hr at the intended cruising point of 27% full power.

As always, the designers were faced with the demands of complex, competing, and inter-related considerations, but to have achieved greater economy with the chosen steam conditions, the cruising turbine would necessarily have had to have been of smaller diameter and operating at a much higher rotational speed.

### **Conclusion**

The improvement in part load efficiency gained by the incorporation of the 'Y100' cruising turbine was ultimately insufficient to justify its complexity. This was particularly the case when engine room internal space and considerations of overall machinery weight came under further pressure with the advent of ship borne helicopter operations and the resulting requirement to include hull hydraulic stabiliser equipment. Similarly, improved weapon and sensors systems required a great increase in the capacity of on-board electrical generating equipment, placing yet further demands on what were already very cramped machinery spaces.

The lack of endurance at cruising speed is attested to by the frequent underway refuelling, Replenishment at Sea (RAS), operations required by the Royal Navy's frigates; necessary not only to increase range but, prior to the introduction of self-compensating fuel systems, also to maintain ships stability when operating long anti-submarine patrols in the North Atlantic.

Issues with the original cruising turbines notwithstanding, the 'Y100' – and derivative 'Y136' & 'Y160' (*Leander*-class) and 'Y124' (LPD Assault Ships HMS Fearless and Intrepid) – steam plants prove to be highly successful and reliable installations. A fitting tribute to the last all-steam warship propulsion machinery designed and manufactured for the Royal Navy.



*A 'Leander' at sea – HMS Juno*

### **Sources**

- 'Review of the Marine Steam Turbine over the last 40 years', D.G. Nicholas;
- 'Recent Development in British Naval Main Propulsion Steam Turbines', F.J. Cowlin & A.F. Veitch;
- 'The Evolution of Warship Machinery 1945 – 1990', Rear Admiral M.A. Vallis, RN;
- 'Machinery Installations of Guided Missile Destroyers and General Purpose Frigates', Commander J.M Dunlop & Commander E.B. Good, RN;
- 'Marine Steam Engines and Turbines', W.J. Fox & S.C. McBirnie;
- 'Steam at Sea', D. Griffiths;
- 'Evolution of Engineering in the Royal Navy, Volume-II', Commander P.M. Rippon, RN;
- 'Leander Class Frigates', R. Osborne & D. Sowdon;
- 'Modern Combat Ships – Leander Class', Commander C.J. Meyer, RN;
- 'British Warship Designs since 1906', G.M. Stephen;
- David Bowie – various correspondences, 1998 – 2010.