Paper read at the Institute on 14 March 1978

# MONITORING AND DIAGNOSING PROCESS DEVIATIONS IN MARINE DIESEL ENGINES

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## SYNOPSIS

This paper describes the basic principals and recent experience with condition monitoring and fault diagnosis in marine diesel engines.

From failure mode analysis and failure distributions it is shown that condition-based maintenance should be the main element in a maintenance system for diesel engines.

A condition monitoring system forms the basis of a condition-based maintenance system. Three different condition monitoring systems are described. A survey of monitoring objectives as well as system lay out and service experiences is given.

In a complex process, such as the diesel engine system, a particular failure is often indicated by changes in several measured parameters. It is therefore necessary for the engineer to have a good knowledge of the system, considerable experience and, very often intuition to determine which failure actually he is dealing with.

A computerbased diagnostic system is described. Both the theory behind the system and service experiences are given.

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## INTRODUCTION

Although in some cases corrective maintenance may be the intended maintenance strategy, the greatest part of all maintenance on a marine diesel engine is directed towards the prevention of failures. On the other hand, within the limits of safe operation, unnecessary preventive maintenance should be reduced or avoided. The need for a better and more cost effective maintenance strategy has lead to the development of condition-based maintenance systems.

The experiences gained during the last 8 years with condition monitoring systems are positive, though there is ample room for further developments. Based on experience with these systems, the Ship Research Institute of Norway has developed a computerized system for diagnosing abnormal process deviations in marine diesel engines.

The continuous sample diagnosis of this system makes it possible not only to expose the real cause of a sudden failure but also to control and follow up gradually developing failures.

## MAINTENANCE STRATEGY

The maintenance concept is defined as the combination of all technical and corresponding administrative actions intended to restore an item to a state in which it can perform its required function.

Rational decision-making in the field of maintenance strategy should be based on analysis of the consequences of different modes of failure and the failure distribution function (reliability analysis). The relevant maintenance objectives are:

- technical condition
- safety level
- costs

The aim is to achieve the functional objectives of an item within cost and safety limits.

The active maintenance actions are generally divided into two different types :-

- corrective maintenance carried out after a serious defect or sudden failure has occurred.

- preventive maintenance carried out with the intention of reducing the probability of reaching the limit.

Further the preventive maintenance can be separated into :-

- scheduled preventive maintenance based on statistically predetermined intervals (mean time between failures)

- condition-based preventive maintenance carried out when quantified pre-described criterias (condition parameters) reach limiting values.

### Failure Distribution

The failure distribution function F(t) expresses the probability of a failure occurring before a time less than or equal to t.

$$F(t) = P(\text{failure before time} = t)$$

By the derivative of this function the probability of failure (instantaneous failure rate) can be expressed

$$f(t) = \frac{dF(t)}{dt}$$

In fig. 1 graphs of the most commonly used models of real life distribution are shown.

a) Exponential distribution.

This distribution is associated with early life failures in the running in period. There is no scope for eradicating early failures by scheduled maintenance because of their nature.

b) Rectangular distribution.

The cause of chance failures is a combination of random circumstances. It is not possible in any rational way to predict the time at which such events will occur and scheduled maintenance of such failures is not possible.

c) Normal Gaussian distribution.

This distribution is used as a model to describe wear out failures. In a mechanical system such failures are the result of time-dependent mechanisms such as corrosion, deterioration or mechanical wear. This distribution was a large standard deviation and makes the scheduled maintenance strategy very unreliable.

Hence, the scope of scheduled maintenance is quite limited, whichever failure distribution is considered; and this is the main reason for the great interest in condition-based maintenance systems.

### Condition-based Maintenance

The decision on when to carry out maintenance is based on measured quantitative information (condition parameter) on the deterioration of components during normal running of the engine.

The condition parameter is established by comparing the instantaneous engine condition with a reference condition (mathematical model) based on a new undeteriorated engine, including an allowance for intolerable deviation which takes into account cost and safety levels. Figure 2 shows a graph of these values versus engine load. The condition parameter should only describe the process deviation due to events and time dependent failures (trend) and should be independent of engine load and surrounding conditions. Unfortunately there will also exist unwanted method deviations caused by simplifications in the mathematical model, measuring accuracy etc.

The table in fig. 3 contains factors affecting the condition parameter. It is difficult to estimate the effect of the different factors but the dependent deviation should not do so much harm. This systematic deviation will occur either as a parallel shifting or as a change in slope of the trend line. It is mainly the random deviation which disturbes the picture. For a parameter assumed to give the trend it is desirable to have a ratio (as great as possible) between the limit value and the standard deviation.

Fig. 4 shows the principal of the trend analysis. The relative parameters from fig. 2 are plotted versus time. This example of trend analysis gives the following information :-

- effect of minor maintenance actions such as servicing, adjustments and cleaning can be visualised by the slope of the trend curve.

- effect of sudden response maintenance actions can be visualised on the curve by the distance C - D.

- effect of process damage can be visualised by the distance A - B.

- prediction of the most likely time of maintenance. The slope of the extrapolated trend line to the intersection with the limit value can be calculated for a selected probability of 50 per cent (mean trend prediction).

## MONITORING OBJECTIVES

Diesel engines used in ships differ with respect to size, specific load, working principle, RPM, etc., which affects their service behaviour. Even if many of the service problems are common, there are also individual problems which must be taken into account when deciding the extent of the condition monitoring system.

### Thermal Load Monitoring

Thermal load monitoring serves two purposes.

a) To prevent cylinder unit damage caused by high metal temperature. This is done by keeping the temperature below a preset limit value.

b) To detect poor combustion and faults in the scavenging process which affect the thermal load. This is achieved by comparing the measured value with calculated reference value. A large difference between these two values indicates errors in either the combustion or the air throughout or in both.

Exhaust Valves.

Burned exhaust valves, caused by high thermal load, are a serious problem mainly for 4-stroke medium speed engines. Monitoring the thermal load of the valves is therefore useful.

The best measure of the thermal load is the temperature on the valve face. However, the temperature is not easily measured on engines in service and an indirect method was therefore chosen.

An investigation carried out on data from several engines showed good correlation between face and seat temperature. The face temperature is obtained by measurement of seat temperature as shown in fig. 5. The measured temperature should not exceed a maximum value above which high temperature corrosion may take place.

Cylinder Cover and Liner.

High metal temperature in the walls of the combustion chamber may cause :-

- cracks in cover and liner due to increased stress caused by high temperature difference between combustion and cooling side.

- high rate of wear of liner and piston rings

- piston seizure

- burning of piston crown

Poor combustion will accelerate the damage processes. Alternative sensor locations are shown in fig. 6.

Combustion Monitoring

The combustion process is the key to engine performance. It affects fuel economy, fouling and wear of cylinder components.

The combustion process is easily disturbed by several factors such as :-

- fuel oil quality
- fuel oil preheating temperature
- fuel oil impurities (water)
- condition of injection equipment

These disturbances express themselves as changes in the following combustion characteristics.

a) Ignition delay

An increase in ignition delay normally means that a large amount of fuel is prepared (i.e. vapourised and mixed with air) before ignition occurs. Thus a large part of the fuel burns in a premixed, uncontrollable combustion. This normally means an increase in the rate of pressure rise and the maximum pressure.

b) Rate of combustion

A low rate of combustion leads to an increased duration of the total combustion period.

A longer combustion period may cause breakdown of the lubrication oil film due to higher liner temperature and a larger part of the liner surface being exposed to the flame, which may in turn result in increased wear of liner and rings.

c) Combustion intensity.

The combustion intensity depends on the rate and character of the combustion.

If the combustion is diffusive in character, radiation comprises a greater part of the heat transfer.

Combustion intensity is best monitored by surface thermocouples in piston crown and cover.

d) Incomplete combustion.

Incomplete combustion leads to the formation of various unwanted products (CO and soot). The result of incomplete combustion is obviously a reduced cycle efficiency. The exhaust temperature is normally not affected.

In order to get a correct picture of the combustion, a number of parameters should be recorded simultaneously. This is possible in a laboratory set up, but in a CM-system on board a ship a selection of parameters must be made which are easy to measure and which have a significant effect on combustion characteristics . In addition to exhaust temperature and metal temperatures etc., dynamic measurements of cylinder and injection pressures are recorded.

Cylinder Pressure Measurements.

Measurement and analysis of cylinder pressures are necessary both for combustion monitoring and for engine adjustments.

From the cylinder pressure diagram in fig. 7 the following parameters are obtained :-

- maximum cylinder pressure

  This parameter is closely related to the length of the ignition delay period and rate of heat release in the initial combustion period.

- cylinder pressure at a fixed crank angle on the expansion line.

  This parameter gives information about after burning and delayed combustion etc. Together with the maximum pressure a rough picture of the combustion process may be obtained.

- ignition timing

  Information concerning ignition delay/ delayed combustion is closely related to this parameter.

- compression pressure

- mean indicated pressure (MIP)

  Equal load distribution between the different cylinders can easily be obtained by measuring the MIP

Injection Pressure Measurement

For a specific engine design the operation depends on how the fuel is injected into the combustion chamber. This depends on the design of the injection system, and also to a very high degree on the condition of the system.

The injection pressure diagram is measured in order to monitor the high pressure fuel pump and injector. Irregularities in the combustion process may also be easier to understand when the injection pressure diagram is known.

The following parameters are derived from the diagram in fig. 8.

- pressure rise before the injector opens

- opening pressure

- crank angle of injector opening

- maximum injection pressure

- injection period (crank degrees)

## Piston Ring Monitoring

Together with combustion and thermal load monitoring, piston ring monitoring completes the condition monitoring of the cylinder unit.

Correct functioning of the piston rings, i.e. ability to prevent gas leakages or blow-by of combustion gases, is vital to the reliability of the cylinder unit.

By using proximity sensors mounted flush with the inner liner wall, it is possible to measure the distance between each ring and the liner surface. A certain positive signal means that the ring is working properly, a low positive signal means that the ring is sticking in its groove or has collapsed. A negative signal means that the ring has broken or disappeared. This detection method requires that the piston rings are made of a magnetic material.

Displaying the signals from the sensors directly on an oscilloscope screen makes it possible to see the condition of the different rings. This of course only gives the instantaneous condition. If the signals were fed into a computer system a statistic treatment of the values over a specified period of time should also be provided.

## Air/Exhaust Gas System

Reduced air flow has been proved to be the cause of many service problems, particularly in highly loaded 4-stroke engines where reduced air flow is the main cause of thermal damage. The air flow should therefore be the main condition parameter in the air/exhaust gas system.

The air flow is affected by :-

- turbocharger efficiency

- the fouling of the system

- temperature of the intake air

A condition monitoring system should comprise the following parameters :-

- efficiency of turbocharger turbine

- efficiency of turbocharger compressor

- pressure drop across air cooler

- pressure drop across engine

- K-value of air cooler

The necessary measurements include air flow, temperature and pressure of air and exhaust gases at various points in the system and cooling water temperature before and after the air cooler.

The pressure drop is measured directly. As pressure drop depends on the air flow, the measured values should be compared to reference values.

As air flow is used in all models, an accurate measurement of the air flow is necessary. At the Ship Research Institute of Norway various measuring methods have been tested out. Fig. 9 shows the following three methods which have given promising results :-

- elbow meter
- diffusor
- Annubar measuring tube

In Appendix C some examples on mathematical models for reference values are given.

## DIAGNOSTIC METHODOLOGY

In a conventional alarm system an engine plant failure is indicated by its response in a single process parameter, and an alarm is given when the parameter value exceeds a fixed limit value. In the complex diesel engine process, a failure in one item very often will give responses in more than one single parameter value. Other failure responses can be detected simultaneously and a number of process parameters should be taken into consideration to expose the real cause of the abnormal condition.

For those parts of the engine process which are too complicated to describe mathematically, it is possible to apply a pattern recognition technique to evaluate a selection of abnormal conditions.

### Normalized Condition Parameter

Fig. 10 shows the principal method for establishing a condition parameter. As seen from the figure the condition parameter is based on measurements taken on the engine in service.

In order to establish the normal parameter a comparison between registered process deviation and comparable, experienced or estimated, process deviation is executed.

The registered deviation can be calculated by the following methods :-

- deviation between a measured single value and the measured/calculated mean value
- deviation between a measured (single) value and the calculated reference value.

The first method can be applied for a large number of equal components. The condition parameter will be independent of engine load if the comparable deviation value is load-independent.

The second is the most commonly used method in the field of condition monitoring. The problem is to establish a good mathematical reference model for a process of high complexity.

The developed diagnostic system was designed for a large-bore diesel engine with 8 cylinders. The mean value comparison was used for the registered deviation and the estimated standard deviation was used as a comparable process deviation.

Assuming that the measured values have a Normal distribution and substituting the mean value as the most expected value, the standard deviation can be expressed as follows :-

$$N = \sum_{i=1}^{n} (p_i - p_{mean})^2/n)^{\frac{1}{2}}$$

For each parameter in the diagnostic system the standard deviation is assumed to be a constant value.

The single value deviation will then be

$$\sigma_i = p_i - p_{mean}$$

If the measured values have a normal distribution then the relative single value deviation

$$p_{i_{rel}} = \sigma_i / \sigma_n$$

will have a standard normal distribution. This means that the expected value is zero and the standard deviation is one.

### Pattern Recognition Technique

From service measurements on the engine plant the measured vector (MV) is established. The condition parameters, CM-parameters, are used as elements in the measured vector.

The diagnosis is based on a comparison between this measured vector and each of a number of pre-defined fault vectors. A pre-defined fault vector consists of condition parameters in a pattern in which they are expected to appear when a certain pre-defined fault really is present. Together they make the fault matrix.

For each pre-defined fault vector (FV) the geometric difference (DV) between this and the measured vector (MV) is calculated as follows :-

$$DV(j) = \sum_{i=1}^{n} (FV(j,i) - MV(i))^2 \text{ for } j = 1,..m$$

n = number of parameters

m = number of predefined faults

The difference vector with the least value is closest to the measured vector. This is assumed to be the most probable fault.

For a system of 3 parameters the method is illustrated in fig. 11. This diagnostic system has 5 pre-defined fault vectors (FV). From the figure it can be seen that the most probable fault is FV(2).

The probability of the actual failure increases by decreasing length of the difference vector. Hence the method is not dependent upon fixed limit values and the sample space of the diagnosis will be continuous.

By following up it should be possible to detect a gradual failure before this develops into a sudden and more serious failure.

## SYSTEM LAY-OUT

### Predikt I.

The concept of condition monitoring by Predikt I was to gain more information from the conventional instrumentation system. New sensors and instruments were also introduced.

The condition parameters, based on manual readings of the instruments, were calculated from nomograms or by desk computer. The parameters are listed in Appendix A.

The following components are monitored without extension of the conventional instrumentation :-

- air filter
- air compressor
- air cooler
- scavenging ports
- exhaust turbine

By the introduction of special sensors and instruments the system has been extended to monitor :-

- thermal load of liner and cover
- piston ring functioning

### Predikt II.

Predikt II is a computerbased condition monitoring system for large bore diesel engines. Condition parameters are listed in Appendix A.

The system hardware consists of :-

- minicomputer (32 K memory)
- tape reader
- tape punch
- Teletype
- multiplexer
- A/D-converter
- fast A/D-converter
- shaft encoder for crank-shaft position information
- pressure transducers for cylinder and injection pressure measurement
- sensors and amplifiers for measuring temperature, pressure, flow, etc.
- proximity transducers for piston ring monitoring

The pressure transducers for cylinder and injection pressure measurements have to be moved from one cylinder to another when taking measurements.

In the Predikt II system no less effort has been put into the lay-out of the man/machine communication. Thus it is very easy to extend both system software and hardware.

### Demos.

Demos is an instrumentation system built for condition monitoring medium speed diesel engines. An outline of the system is shown in fig. 12 and it is primarily designed for online measurements, but due to a flexible construction, a broad range of variations in both measurements and presentation methods can be obtained.

The minimum system configuration, designed for measurements and analysis of the combustion process in a diesel engine, consists of :-

- pressure pick-up (including amplifier) which may be connected to the respective cylinder
- shaft encoder for crank-angle position information
- fast A/D-converter
- minicomputer
- operators control panel consisting of push buttons for control actions and alpha-numeric display for presentation of results.

It is possible to extend the minimum system configuration in many ways, both with regard to instrumentation, analysis functions and presentation facilities. A full size system will include the following functions :-

- continuous combustion monitoring. Pressure sensors are permanently installed on all cylinders for measurements of cylinder pressures.
- injection system monitoring. The injection pressure is measured and analysed either by means of a set of permanently installed pressure sensors or a single portable sensor.
- monitoring of specified measurements, i.e. bearing temperatures, scavenging air pressure etc.
- thermal load monitoring of cylinder covers and liners.
- alarm reporting. All measured and calculated values are checked against alarm limits.
- plotter for presentation of measured and computed sequences.
- presentation on hard copy.

## SYSTEMS IN SERVICE

### Predikt I.

The first Predikt I system was delivered at the end of the sixties. The system was designed for condition monitoring of large bore 2-stroke diesel engines.

The experience with the "manual" Predikt I system has not developed as planned. There are several reasons for this.

- the amount of work involved in carrying out a condition test has been too great. This is mainly due to the use of nomograms. A programable desk computer has solved this problem.
- lack of information about the system provided for the user has certainly caused repugnance towards the system due to the work involved.
- a large scatter in the condition parameters when plotted versus time. Due to random deviations caused by variations in the process itself (non-simultaneous readings) and inaccuracies in reading the instruments and/or using the nomograms, the prediction of maintenance actions has not been successful.

The section of the Predikt I used for thermal load and piston ring monitoring (thermal load analyser and piston ring oscilloscope) has been quite successful. This is due to the following :-

- this part of the system is easy to operate
- the signals from the thermal load analyser and piston ring oscilloscope are easily interpreted.
- the signals give direct information about the actual condition of the cylinder unit.

## Predikt II.

The work with the Predikt II system started in 1969 with a test installation on board the OBO-carrier M/V "VIANNA". The system was fitted onboard the ship for 2.5 years. All the time research engineers from the Ship Research Institute of Norway were onboard. The purpose of this installation was to :-

- establish and verify condition parameters of different components
- test hardware in marine surroundings
- test software for data-acquisition and data reduction

In 1975 the Ship Research Institute of Norway delivered a Predikt II system to the Institute for Mechanical Constructions (TNO) in Holland. The system was installed onboard the dry cargo ship, M/V "TRIDENT AMSTERDAM", which is powered by a SULZER 8 RND 76 engine.

The peripherals (punch, photoreader etc.) have been functioning very satisfactorily during the two years of operation. Only the Teletype has had one failure.

The computer itself has, however, failed several times. In most cases the faults could not be traced by the standard diagnostic programmes, and a "trial and error" search for the source of malfunction had to be used. As a last effort to put an end to the problems all prints in the computer were renewed by the manufacturer in July 1977. Up to now no problems have been reported.

The Ship Research Institute of Norway has seven years of experience with this kind of computer, including in marine surroundings, with very good results. The experience onboard "TRIDENT AMSTERDAM" must therefore be considered as a "worst case".

The first problems occurred with the proximity transducers for piston ring monitoring. All transducers were replaced by a redesigned type which is still working properly.

Other sensors and interfacing equipment (amplifiers, measuring bridges etc.) have performed quite well. Minor problems have been solved by the ship's staff.

Metal temperatures were found to fluctuate widely and from continuous measurements of metal temperatures it was found that :-

- unstable phenomena occur in the cylinder even at constant engine load.
- the frequency of the instabilities is irregular

The reason is thought to be due to irregular ring travel. Thus, from this data the detection of a trend is not possible.

For the parameters connected to the air/exhaust gas system good results have been obtained. Fig. 13 shows the variations in turbine efficiency.

The trend development for each condition parameter is plotted versus time on a graphical display. Cylinder pressure and injection pressure diagrams may also be shown on the display.

The signals from the proximity transducers are displayed on a special oscilloscope screen. Fig. 14 shows the signal from one of the sensors.

The experience with this kind of measurement has been very good. Practical results have coincided with theory.

## Diagnostic System

The diagnostic test programme in the Predikt II system started in November 1977. The failures the programme is able to detect and the programme input are listed in Appendix B.

The testing consisted of introducing failures into the engine but due to the limitations for safe operation of the engine and ship, only minor failures could be introduced :-

- spill valve of h.p. pump, cylinder no. 5, was adjusted to open too early.

- injector spring of cylinder no. 2 was slackened.

On the printout the five most probable faults for each cylinder are printed. For cylinders nos. 2 and 5 the introduced failures came out with the highest probability. The printout is partly shown in fig. 15. When measuring the engine and running the programme 24 hours later the same result was achieved.

Moreover, after removing the introduced faults, the programme came out with the same readings. The reason for this is assumed to be :-

- when correcting the introduced faults, the situation was not brought back to original condition

- the inaccuracy of the injection pressure measurement

It was found that the measurement of injection pressure diagram was not good enough. Due to sampling only once per crank degree, the opening pressure, crank angle of opening pressure and thereby the injection length could not be found with sufficient accuracy. Tc obtain the required accuracy the sampling must take place at least every half crank degree or better still every quarter of a crank degree over the period of injection.

The diagnostic programme will be further tested both onboard the ship and in a laboratory engine during the coming year.

Demos.

The prototype system, which is identical to that shown in fig. 12, was designed and built at the Ship Research Institute of Norway. The work started early in 1974 and the system was ready for sea in May the following year.

The system was installed onboard a Norwegian ship with medium speed propulsion machinery (2 x MAN V6V, 40/54) during the second half of May 1975. Only one of the engines was fitted with the system.

The aim of the prototype installation was to test monitoring methods as well as hardware and software in general. During the 18 months long test period valuable experience was gained.

Except for one memory error the computer has been very satisfactory and the actual computer seems well suited for shipboard use.

Some of the peripherals caused problems, especially during the first 12 months when some of the components were modified and one was replaced. This seems to have solved the problem as no failures were experienced during the next 6 months.

Two types of sensor for continuous measurement of combustion pressure were used. One of them worked satisfactorily all the time the other failed rather frequently during the first 6 months. After certain modifications by the manufacturer, no failures occurred during the rest of the test period.

Standard sensors are used for all low frequency measurements. The way of handling the signals is, however, not standard. In order to get a simple and less expensive system no amplifiers are used for temperature measurements. This is rather unusual as signal amplifiers are regarded necessary due to noise from other electrical equipment.

During the test period no problems at all were experienced as far as temperature measurements were concerned.

A characteristic problem for highly loaded medium speed engines is to get an even load on each cylinder, adjustments usually being done by adjusting the fuel pumps to get even maximum cylinder pressures and exhaust temperatures.

During the test period the engine was totally overhauled and tuned in the usual way by specialists, and over the next few months results showed that :-

- MIP varied about 9 per cent between cylinders
- thermal load varied about 15 per cent between cylinders but the exhaust temperatures remained constant
- ignition angle varied more than 2 crank degrees

The experience gained during the 18 months long test period can be summarized as follows :-

(1) The conventional instrumentation did not give the engineers sufficient information about the condition of the engines. Additional instrumentation, like Demos or similar, is necessary to achieve economical and safe operation.

(2) Most of the service problems on medium speed engines are caused by low air throughput and poor combustion. Additional instrumentation should therefore be based on cylinder pressure analysis and measurement of air throughput.

## CONCLUSIONS

With the above background knowledge and due to the considerable variations in load, internal and external conditions of a diesel engine, there is still insufficient knowledge to produce accurate mathematical models of diesel engine systems and components. The success of any data acquisition and analysis system depends on the accuracy and reliability of the information the system is based on.

Unfortunately, mathematical models will always contain unwanted deviations, caused by the measuring methods used as well as inaccuracies, which are either uneconomic or impossible to remove.

However, recommendations for the installation and scheduled calibration of instruments and transducers have to be given a very high priority in engineering analysis.

Although the simple Predikt I system is encumbered with failures and inaccuracies, used with prudence, it can be a valuable aid to get maximum information from conventional instrumentation.

Instrumentation of a computerbased condition monitoring and diagnosing system has to be economically justified compared with the expensive Predikt I system. Multipurpose use of the computer would make it more cost-effective, and the future of computer systems onboard ships lies here. The accuracy of a computerbased system will improve with increased measuring accuracy (special and better instrumentation), no influence from the process variations (simultaneous measurements) and more complex mathematical models.

The condition monitoring system should not be allowed to replace the engineers' experience, intuition and understanding of system behaviour, but should be a supplementary to these human qualities. The main goal is to increase the information level upon which the engineer makes his decisions, and thereby increase the probability of the decision being correct.

The Ship Research Institute of Norway is now running a new project which will make a further analysis of the data obtained with a view to attempting to define the mechanism of failure and investigate the reliability of the marine diesel engine.

## APPENDIX A

Condition Parameters for Predikt I :-

1. Air filter pressure drop
2. Compressor efficiency
3. Air cooler thermal conductivity
4. Air cooler pressure drop
5. Engine pressure drop (scanvenging air pressure)
6. Exhaust turbine efficiency
7. Metal temperature cylinder cover
8. Metal temperature cylinder liner man. side
9. Metal temperature cylinder liner exh. side
10. Piston ring functioning

The standard Predikt I system now only monitors parameters 7 - 10.

Condition Parameters for Predikt II :-

1. Metal temperature cylinder cover
2. Metal temperature cylinder liner man. side
3. Metal temperature cylinder liner exh. side
4. Air flow
5. Scavenging air pressure
6. Compressor efficiency
7. Exhaust turbine efficiency
8. Air cooler thermal conductivity
9. Air cooler pressure drop
10. Air filter pressure drop
11. Piston ring functioning
12. Cylinder pressure diagram
13. Injection pressure diagram

Condition Parameters for Demos :-

1. Air cooler thermal conductivity
2. Air cooler pressure drop
3. Compressor efficiency
4. Exhaust turbine efficiency
5. Metal temperature exhaust valve
6. Cylinder pressure diagram

APPENDIX B

Faults Detectable by the Diagnostic Programme :-

1. Normal condition
2. Injection timing late
3. Injection timing early
4. Injection nozzle blocked
5. Injection nozzle wear
6. Injection spring tight
7. Injection spring slack
8. Injection pump plunger wear
9. Late opening of spill valve
10. Abnormal liner wear
11. Piston ring broken or collapsed
12. Piston top deposit
13. Scavenging port fouling
14. Air flow low
15. Early opening of spill valve
16. Fuel oil temperature low
17. Late closing of suction valve
18. Crank in cylinder component

Input Parameters to Diagnostic Programme :-

1. Cyl. mip
2. Cyl. compression pressure
3. Cyl. max. pressure
4. Cyl. expansion pressure
5. Inj. opening pressure
6. Inj. max. pressure
7. Inj. angle
8. Ign. angle
9. Ign. delay
10. Inj. length
11. Exh. temperature
12. Metal temp. cover
13. Metal temp. liner man. side
14. Metal temp. liner exh. side
15. Excess air
16. Spec. air consumption
17. Spec. fuel consumption

APPENDIX C

Some mathematical models for reference values :-

Metal Temperature Cylinder Cover and Liner

$$t_{REF} = C_1 + C_2 \cdot MIP^{Z1} \cdot T_S^{Z2} \cdot P_S^{Z3} \cdot N^{Z4} \quad \text{(I)}$$

Pressure Drop Across Air Cooler

$$P_{REF} = C_3 \cdot L^{Z5} \cdot T_c/P_c \quad \text{(II)}$$

Air Cooler Thermal Conductivity

$$K_{REF} = C_4 + C_5 \left(1/\left(\frac{1}{L^{Z6}} + \frac{1}{S^{Z7}}\right)\right) \quad \text{(III)}$$

Maximum Cylinder Pressure

$$CMAX_{REF} = C_6 + C_7 \cdot (MIP \cdot N)^2 \quad \text{(IV)}$$

Maximum Injection Pressure

$$PIMAX_{REF} = C_8 + C_9 \cdot N \quad \text{(V)}$$

Notation

$C_1$, $C_2$, ......., $C_9$ are empirical constants

Z1, Z2, ......., Z7 are empirical constants

MIP = Mean indicated pressure

$T_S$ = Scavenging air temperature before cylinder

S = cooling water flow to air cooler

N = Engine RPM

L = Air throughput

$T_c$ = Air temperature across air cooler, mean value

$P_c$ = Air pressure after air cooler

$P_S$ = Scavenging air pressure before cylinder

S = Cooling water flow to air cooler

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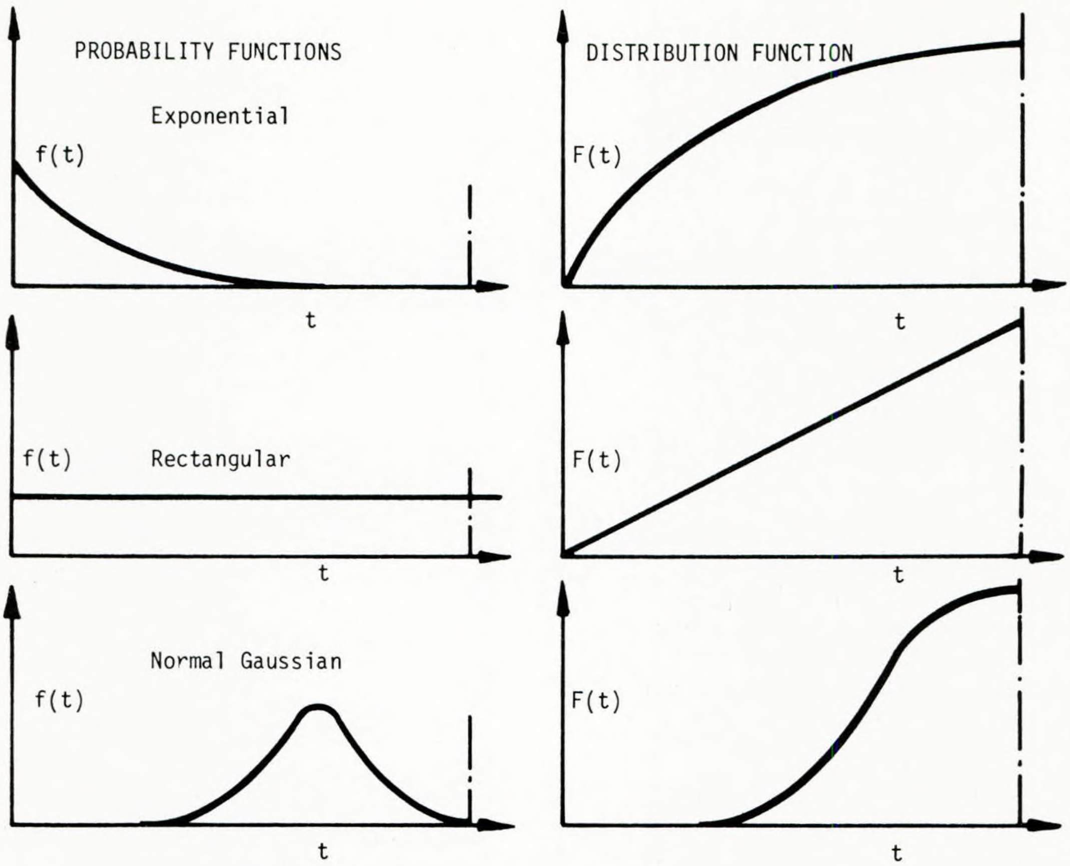


Figure 1. Failure Distribution.

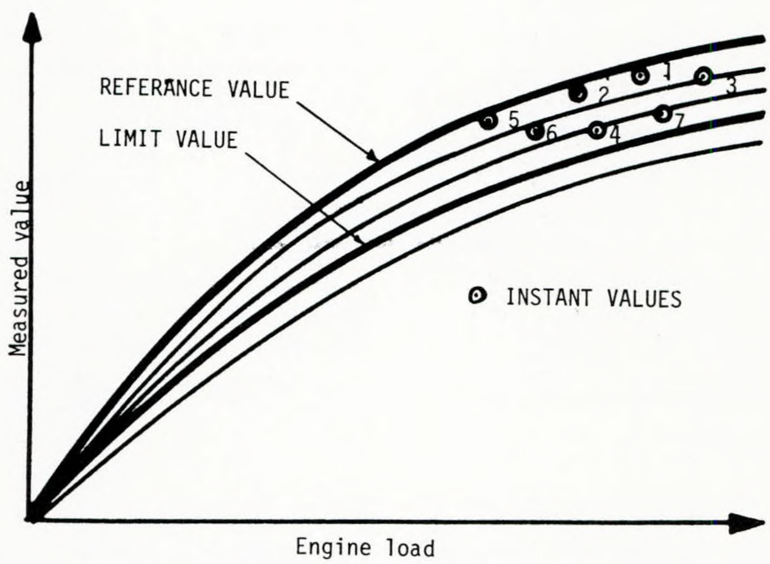


Figure 2. Parameter Characteristics.

| UNWISHED PROCESS DEVIATIONS WHICH WE TRY TO DETECT | | UNWISHED METHOD DEVIATIONS WHICH WE TRY TO CONTROL | |
|---|---|---|---|
| TREND | EVENTS | DEPENDENT | RANDOM |
| Wear | Maintenance | Measuring method | Measuring accuracy |
| Fouling | Breakdown | Meas. equipm. position | Process variation |
| Corrosion | Damage | Mathem. model | Mathem. model |

Figure 3. Factors influencing tne Deviation.

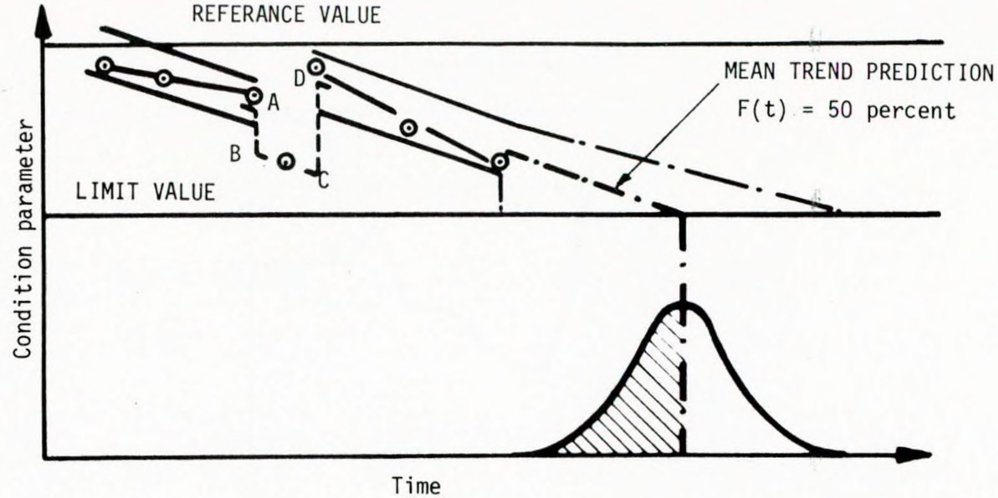


Figure 4. Trend Analysis

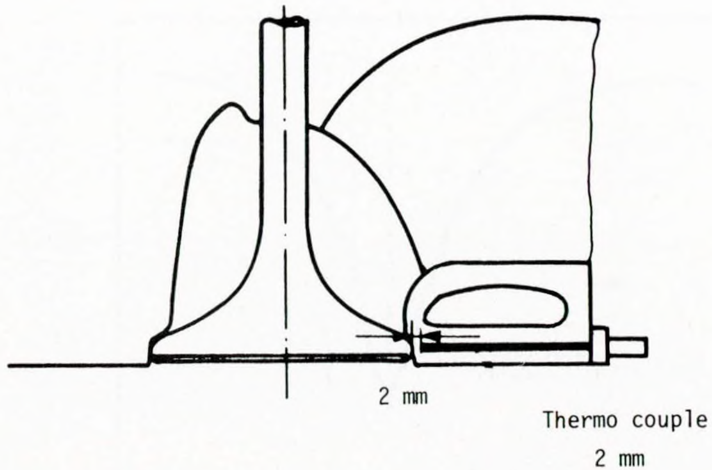


Figure 5. Measurement of valve seat temperature.

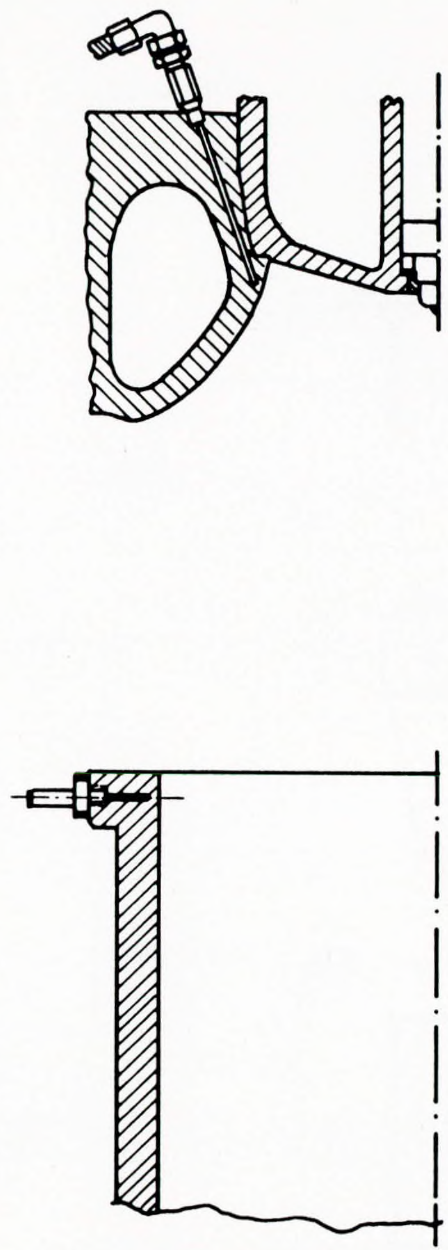

Figure 6. Location of Thermo Couple in Cover and Liner.

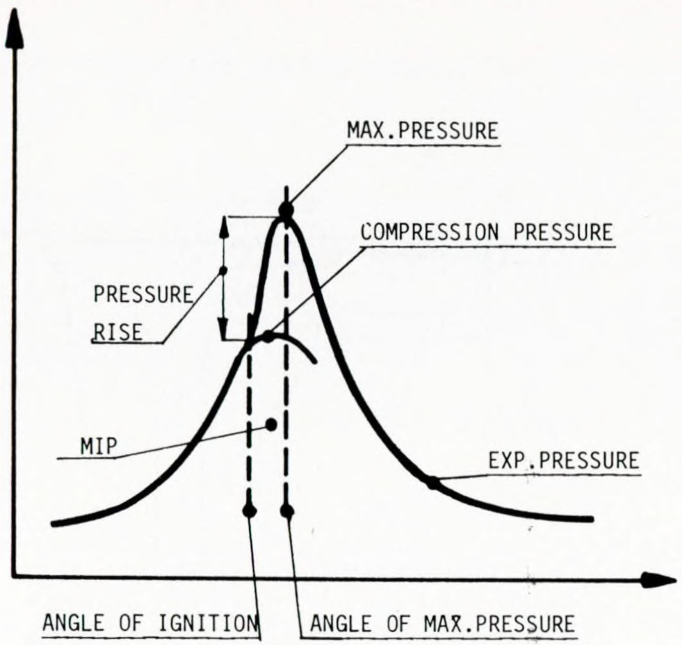


Figure 7. Cylinder Pressure Analysis.

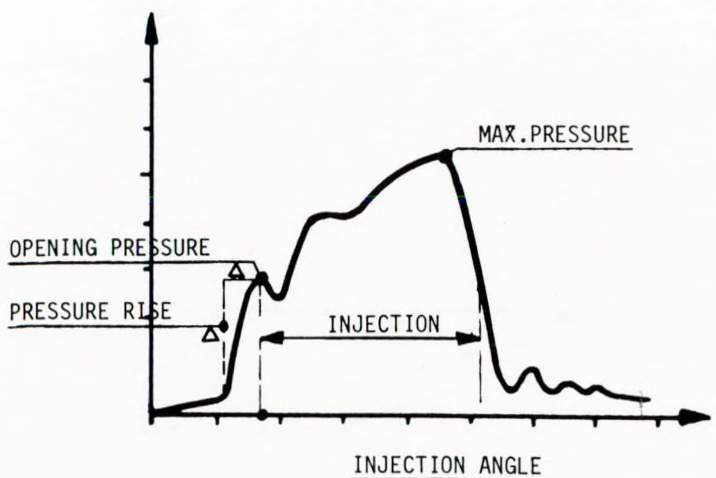


Figure 8. Injection Pressure Analysis

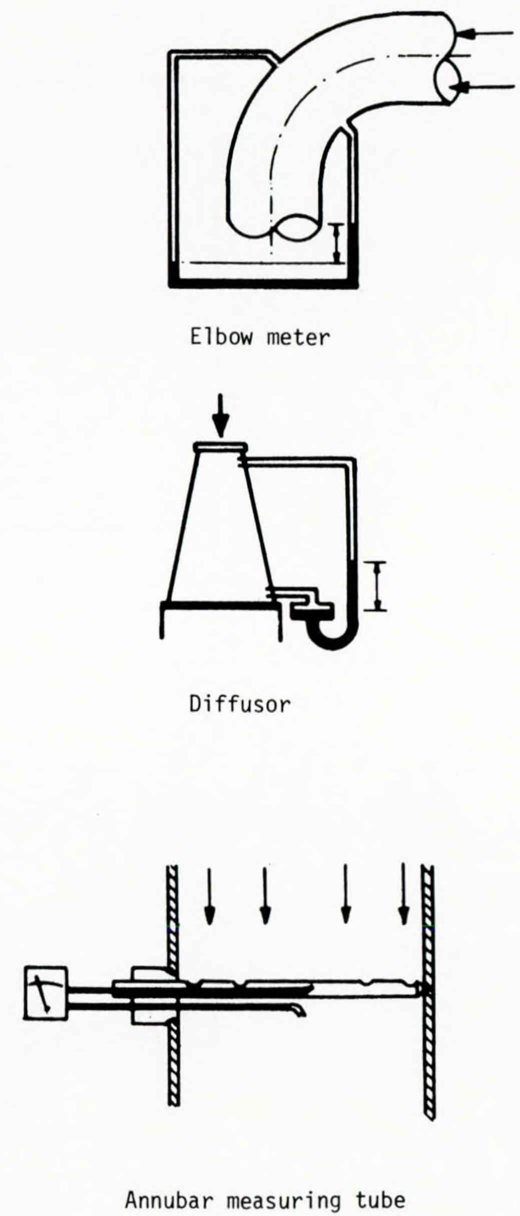


Figure 9. Air Flow Measurements

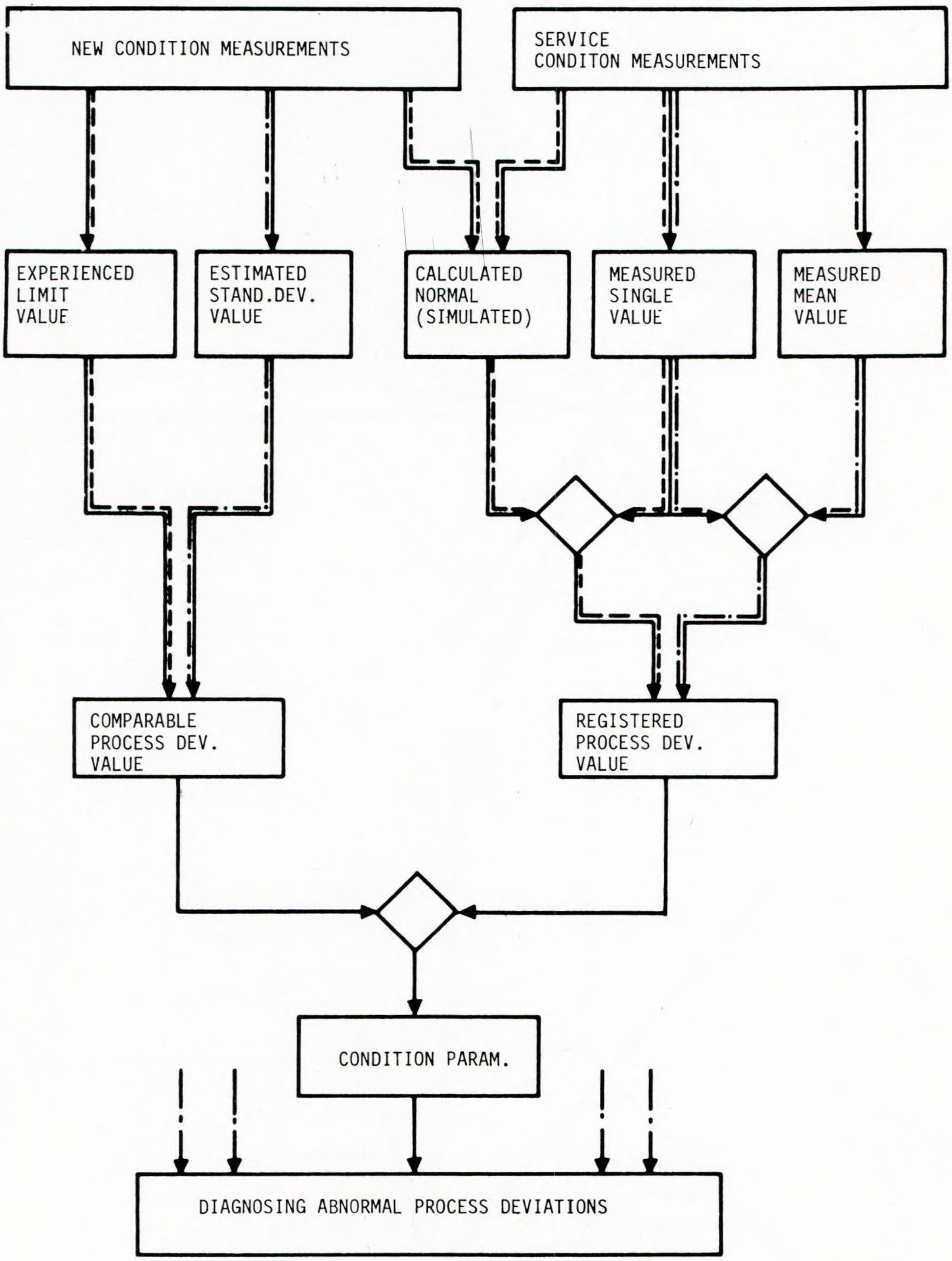


Figure 10. Calculation of Condition Parameter.

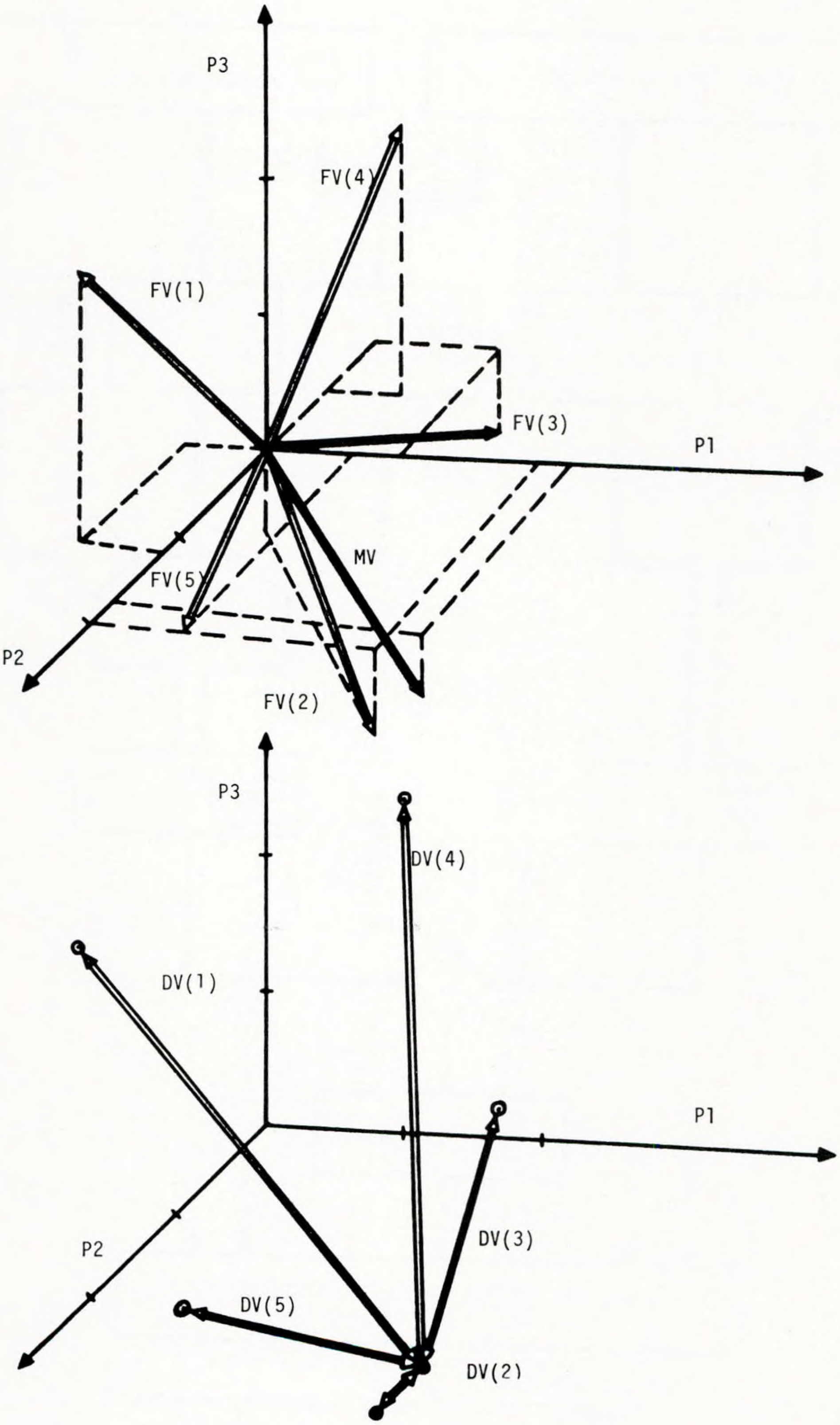


Figure 11. Principal of Diagnosis.

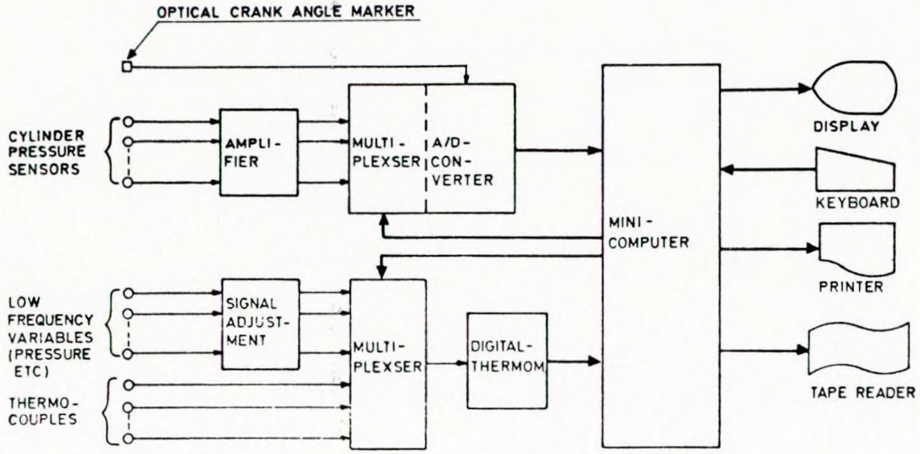


Figure 12. Demos Configuration

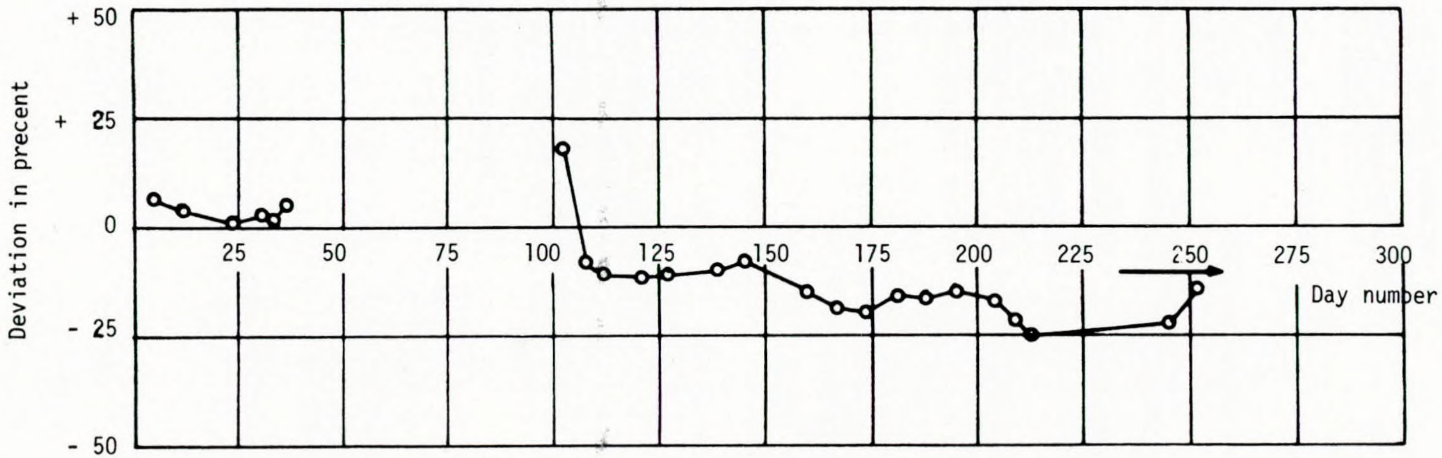


Figure 13. Turbine Efficiency.

*** DIAGNOSIS OF CYL. NO: 2 ***

MEASURED VECTOR:
1 1 1 1 -2 -2 0 -1 -1 1 0 0 -1 0 -1 -1 -1

POSSIBLE FAULTS RANKED BY PROBABILITY:

| FAULT NO. | FAULT DESCRIPTION | VEC.DIFF | PROBABILITY |
|---|---|---|---|
| 7 | INJECTION SPRING SLACK | 7.36 | 86.761 |
| 1 | NORMAL CONDITION | 29.30 | 56.808 |
| 12 | PISTON TOP DEPOSIT | 41.27 | 45.091 |
| 9 | LATE OPENING OF SPILL VALVE | 45.83 | 41.287 |
| 3 | INJECTION TIMING LATE | 52.42 | 36.354 |

*** DIAGNOSIS OF CYL. NO: 3 ***

MEASURED VECTOR:
0 0 0 0 0 0 0 0 1 1 1 0 0 0 0

POSSIBLE FAULTS RANKED BY PROBABILITY:

| FAULT NO. | FAULT DESCRIPTION | VEC.DIFF | PROBABILITY |
|---|---|---|---|
| 1 | NORMAL CONDITION | 9.34 | 72.825 |
| 9 | LATE OPENING OF SPILL VALVE | 20.88 | 49.240 |
| 13 | SCAVENGING PORT FOULING | 23.99 | 44.311 |
| 7 | INJECTION SPRING SLACK | 32.52 | 33.175 |
| 12 | PISTON TOP DEPOSIT | 33.29 | 33.009 |

*** DIAGNOSIS OF CYL. NO: 5 ***

MEASURED VECTOR:
-2 0 0 -2 1 0 0 0 0 -3 -3 -1 0 -3 0 0 -1

POSSIBLE FAULTS RANKED BY PROBABILITY:

| FAULT NO. | FAULT DESCRIPTION | VEC.DIFF | PROBABILITY |
|---|---|---|---|
| 15 | EARLY OPENING OF SPILL VALVE | 12.30 | 86.165 |
| 1 | NORMAL CONDITION | 63.75 | 46.228 |
| 5 | INJECTION NOZZLE WEAR | 66.20 | 44.880 |
| 8 | INJECTION PUMP PLUNGER WEAR | 68.66 | 43.564 |
| 2 | INJECTION TIMING EARLY | 76.64 | 39.552 |

Figure 15. Print—out from Diagnosing Program

DISCUSSION

MR J L BUXTON BSc MIMarE (Lloyd's Register of Shipping), said that in their analysis of diesel engine parameters the authors had concentrated on monitoring the combustion process and thermally loaded parts of the engine. It would appear that no account had been taken in their diagnosis of the effects of hull fouling which would effect the ship's speed and the rev/min of the propeller.

If ship speed was maintained then the engine might become thermally overloaded with the resulting possibility of cracked liners. This was a rather more common mode of liner failure than was wear down, which received so much attention from the monitoring point of view. Should not the condition monitoring diagnostic system be able to detect a loss of ship speed and propeller rev/min and distinguish why it was occurring so that the thermal and mechanical loading of the propulsion machinery might be optimised ?

It would also be of interest to learn if any account was taken on the effects of hull deformation. Deformation of hull structures by way of transverse framing and engine seatings could affect bedplate and bearing alignment and could also affect gearing flexible couplings. Was any attention paid to the subsequent wear and heating effects in these areas ?

Finally, the installation of a condition monitoring system represented additional equipment which was liable to failure and hence required maintenance. The authors had pointed out that the computer in the Predict II system had failed several times. Even allowing for the experimental nature of the project, could these faults be diagnosed and corrected by the ship's staff ? If this was not the case and specialized engineers were required to correct each fault, then it was possible that the system would rapidly detiorate. With regard to sensor faults, was it possible to change defective sensors without affecting the running machinery, and did a defective sensor ever give rise to a machinery fault ?

COMMANDER E M S WINDRIDGE RN (Royal Naval College Greenwich) had been most interested to read of the way Ship Research Institute dealt with the information derived from the monitoring system. (These were the processes illustrated in Figures 10 and 11 of the paper).

At ICMES 1977 Paris Conference on condition monitoring, Professor Kawasaki* read a short paper on this particular aspect of the process. He drew a firm line between monitoring for preventive maintenance and statistics for scheduling periodic maintenance in a number of identical machinery systems. That is, between the problem of the individual ship operator, motivated by the need for effective maintenance of his own ship and the planning of ship maintenance/availability by fleet superintendents. The authors of the present paper and Professor Kawasaki were agreed on the need for adequate mathematical models and the need to be able to deal with probablistic data in the monitoring system. However, Professor Kawasaki further stated:
"The principle of monitoring to predict maintenance is to take preventive actions by a monitor which predicts failure deterministically".

Was the proposition that the diagnostic process described in Figure 11, (variance between the predicted vectors for failure and the monitored vector), actually a measure of failure probability, or was it a process dealing with a distribution of input data ?

It would appear that there was also an assumption that the vector described as DV in Figure 11 would not change direction as the predicted and monitored vectors approached. If, for example, $P_3$ varied much more rapidly than $P_1$ or $P_2$ then the measured vector would would curve away from its course towards FV (2) conceivably finishing up at FV (5). Was the assumption that the measured parameters varied at approximately the same rate justified in practice ?

Was it the intention to supersede planned maintenance scheduling by condition monitoring, or was condition monitoring to be regarded as an adjunct to scheduling, supplying additional information which justified modification of maintenance requirements ?

Many diesel operators had great confidence in scheduled maintenance systems. Should condition monitoring of individual engines be associated with modification of fleet schedules ?

*Kawasaki, Y 1977 "A Quantitative Analysis and Evaluation of the Monitored Maintenance Applied to a Shipboard System" ICMES Conference on Conditioning Monitoring and Preventive Maintenance. May.

MR W R O MANN FIMarE (BP Tanker Company Limited), in a contribution read by Mr Thomas, congratulated the authors on sharing the results of their comprehensive look into an area of condition monitoring which was both topical and promised to be a growth area. They had done much useful work in investigating some of the less well recognized parameters of the diesel engine system.

However, Mr Mann wished to make several points:

1) For any ship installed system or technique which had innovatory elements it was basic to the success of the enterprise that the ship's staff were involved to a degree whereby they identified with it, and were as keen on its success as the promoters. Predikt I, for example, caused 'repugnance' - this reaction echoed his company's own experiences in similar situations.

2) Their approach to developing a ship based condition monitoring system had been to evolve the existing functions of the watchkeepers, basically in directions dictated by service difficulties. Instead of using mathematical models they had drawn on a wealth of experience and expertise within their fleets.

With regard to the diesel engine, both main and auxiliary, the areas identified had been cylinder conditions and the maintenance of air cooler/ turboblower efficiency.

3) For cylinder conditions, his company's efforts were concentrated in the Servodyne engine monitor which gave an oscilloscope display of combustion and fuel pressures.

The visual presentation was consistent with the involvement of ship's engineers mentioned earlier. It had been found that they could readily interpret the display and with a little experience could quickly spot inconsistencies and poor conditions.

The use of a computer probability analysis in this area of operation did not seem necessary. The next step the company could see in this field was the ability to adjust fuel pump timing whilst the engine was operating; a facility not easily attained on present engines. In this way the dynamic feature of the oscilloscope display was fully utilized and the management of the engine was firmly under the control of the manager.

This dual potential of visual display and timing adjustment assumed greater significance as the fuel quality inevitably declined.

With regard to the air/exhaust system he would agree with the authors that some form of trend analysis must be employed. Effort was needed to acquire reliable data upon which to base the analysis.

The company's experience with liner wear and piston ring monitoring had not been good, mainly due to equipment failures. Much valuable information was available here and one hoped that, in time, the transducers and allied equipment would be produced to handle the requirement.

As a general comment he would again agree with the authors that current levels of instrumentation were inadequate in accuracy, reliability and scope, if a condition monitoring system was to be applied. It was pleasing to note that the major large bore diesel manufacturers were involved on their own account with systems similar to the one described this evening.

The auxiliary diesel engines were apparently not receiving the same attention.

The equipment described in the paper would appear to be aimed at new construction. Mr Mann would again agree that for the future such condition monitoring systems would be necessary to adequately supervise the invested capital. Did the authors see an application for existing tonnage ?

Finally, the other element he saw as needing careful development was the interface of the condition monitoring system and the ships' operators; the company's policy had been to retain them as the decision makers, and the transition to computer type probability terminals would need to be handled with care.

MR G VICTORY FIMarE said that although the paper might show something of the shape of things to come, it did not yet represent a practicable and reliable operational system for commercial vessels at this time. The authors had examined the commonly used models of exponential, rectangular and gaussian fault distribution in machinery. They had said that the chance of fault detection or eradication by scheduled maintenance had no scope; was not possible; and was very unreliable. This led to a general statement that "the scope of scheduled maintenance was quite limited". This was open to doubt, for the much maligned "scheduled maintenance" had been the basis for the satisfactory running of the marine diesel engine for over 50 years. Perhaps it was wasteful of material and labour, but many of these old engines ran reliably year after year. Perhaps the modern diesel engine did not have the long running capability of the older, less highly rated engines. Many breakdowns were a result of a breakage or material fault in some part of the engine (rectangular distribution) and such failures, together with many of the "early life" failures (exponential distribution) were equally unlikely to be detected by the diagnosing system proposed.

The concept relied on the availability of a great deal of information not generally available to Chief Engineers, and this had to be provided reliably and continuously by expensive and complicated instrumentation. Despite comments as to its reliability in the tests and trials carried out by the author's organization, most marine engineers, because of some unfortunate experiences of such aids, would need more convincing of the continued functioning of instruments and electronic equipment under normal operational conditions. Perhaps if the operation of the machinery was monitored as proposed and the information presented to the Chief Engineer instead of to the computer, he would be in a position to diagnose any faults without the additional cost and complication of the electronics and computer diagnosing system.

It was difficult to see how some of the variables which could affect various engine parameters, i.e. weather conditions, hull fouling and change of fuel quality, could be fed into the diagnosis system suggested, whereas the Chief Engineer would be aware of these conditions at all times and would know how to correct for such variables in his diagnosis of the machinery under his control. It was hardly realistic to say that "the condition monitoring system should be supplementary to the engineers' experience, intuition and understanding of system behaviour" for this ignored the possibility of a conflict between the Chief Engineer's considered opinion and the probable fault indicated by the monitoring system. In such a case the Chief Engineer would be in an impossible position - he would be criticized if he acted according to his conviction if this was wrong, and also if he acted according to the system diagnosis if this was shown to be at fault.

The ability to feed realistic "Mean Trend Predictions" into such systems depended on obtaining reasonably uniform information relating to as large a number of similar engines as possible, and was based on the assumption that similar engines had a similar wear deterioration pattern. This was a doubtful starter on two counts,
i) there were not many identical marine engines built in any one series and
ii) so called "identical engines" did not wear or deteriorate in a uniform manner. To expect them to do so would be akin to expecting all cars of one make and model to give similar service, or for any one of a group of people to react in the same manner when subjected to similar pressures and stresses. As most marine engineers would confirm, similar engines could not be expected to exhibit similar characteristics, wear rates, or failure patterns. Like motor cars and human beings, they did not and would not!

CORRESPONDENCE:

MR H D MAKINSON wrote to say that the paper had indicated most interesting trends with regard to monitoring diesel engines under seagoing conditions. It showed, amongst other things, the necessity of giving much greater attention to this type of instrumentation in present and future training programmes for both marine engineers and marine engineer cadets.

Was it possible for the authors to give a little more detail with regard to the pressure transducers used ? Were the transducers of the "piezo type", and were they water cooled or air cooled ?

Inevitably modern instrumentation brought with it both the bane as well as the benefit of today's sophistication, particularly with regard to cost. A water cooled piezo pressure transducer cost £488 plus VAT. Capital costs for this equipment as used in training were high and, at present, there was not, in general, the will to face this. As damage through dampness or overheating could all too easily occur, it appeared that the development of a much cheaper type of reliable transducer was necessary and long overdue.

The most essential ingredient in the monitoring was very reliable equipment, because in the pursuit of reliability and uninterrupted service for ships, the comment of an eminent shipowner and past President of the Institute, of some years ago, should not be forgotten: "Ships are not laboratories tended by single minded specialists who can concentrate on their own particular problem". With reduced manning today, all equipment needed to be rugged enough to be reliable in the hands of the non-specialist user.

Mr Makinson hoped the authors could give further results in the future course and, if possible, give more detail on the type of transducer in the fuel delivery line.

DR R A COLLACOTT FIMarE wrote that diagnostic monitoring as set out by the authors was a developing technology which offered very considerable returns to shipowners and others involved in the operation of capital plant. During the last three years, knowledge in this field had extended considerably, ranging from the use of computer-assisted diagnostic techniques in space-flight, aviation and power generation to the development of an "Anticipator" in process industries as stimulated by the

Flixborough disaster through to a self-integrity appraisal of fire-alarm systems to reduce spurious alarms[1 2 3].

It was interesting that the authors should introduce the need for failure mode analysis since this was dealt with in an excellent paper to the Institute by Davies[4] and by Venton and Harvey[5] from Y-Ard, Limited and such procedures remained an important pre-design process. In the paper presented it would appear that the aim upon which the whole project was based was that of maintaining high performance throughout working life and for this reason the monitors, as developed, concentrated on providing continuing combustion performance. Under different conditions the aim might well be related to safety and with such objectives the monitoring and diagnostic techniques would possibly be applied to other system components, thus in the aircraft industry, monitoring based on performance and extended from the "black box" crash recorder made more use of vibration monitoring and of "deltas" in thermodynamic deviations[6 7]

This paper had appeared at a significant time when the full possibilities of condition monitoring were being explored and adopted by other industries. Condition monitoring involved a high degree of technology in the introductory phases but once commissioned could be self-compensating and provided protection and maintenance with a minimum of attention - exactly the sort of condition suited to marine engineering and other hazardous occupations.

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7) FELTHAM R G 1970 "The role of the flight recorder in aircraft accident investigation" R Roy Aero Soc 74 No 7 pp 573-576

MR E F KIRTON wrote that the authors' forthright reporting of their failures as well as their successes was impressive and did them credit. He realized that the achievements had not been gained cheaply and asked whether the work had been financially supported by shipbuilders, owners or manufacturers. Also, if the authors did not consider the question too impertinent, what was their estimate of the effort, both in cost and many years work, that was attributed to the project.

The unstable nature of the cylinder metal temperature under steady operating conditions referred to by the authors had also been reported by Mr J F Butler in his paper on the Doxford/Hawthorn Seahorse engine* and was attributed in that paper to the rotation of the piston rings. Since the temperature variation appeared to be confined within determinable limits whilst the rings remained healthy, Mr Kirton asked for the authors views on the use of thermocouples as an alternative to proximity transducers for monitoring piston ring condition.

* IMarE Transactions Part 3 1975

MR A TROUSSE (Bureau Veritas, Paris) wrote that as a Classification Society, Bureau Veritas was obviously interested in the advanced techniques of diagnosis processing and also of predictive maintenance, as to ships in service.

The Society would also like to learn more about the credibility of the interesting results gained from this attractive method and system. In particular, as they had always placed great importance on the accuracy and reliability of instrumentation in the scope of their automation mark AUT, the Society would like to obtain further explanations and assurances about the quality of both the mathematical models and instrumentation which had enabled the authors to state that MIP varies about 9% and thermal load about 15% between cylinders, while exhaust temperatures remain constant.

In addition, did the authors consider that this new monitoring method should be a necessary complement to the classical method of surveying the exhaust gas temperatures of diesel engines fitted in unattended machinery rooms ?

AUTHOR's REPLIES

The authors thanked the contributors for their interest in their work; this discussion had become a valuable addition to their paper.

They fully agreed with Mr Buxton that the monitoring of hull fouling should be a part of the condition monitoring system. The Ship Research Institute of Norway had worked with this problem for some time, and monitoring of hull fouling was also a part of the Data Trend system developed in co-operation with Norcontrol A/S.

However, this part of the condition monitoring system had not yet been very successful, due to the problem of measuring the correct ship speed. Even the new types of ship logs had not the necessary accuracy.

Concerning the effect of hull deformation on engine and bearing alignment and thus heating effects in bearings, gears etc: this had not been taken into consideration in their work.

The Ship Research Institute of Norway had so far mainly concentrated on the condition monitoring and diagnosing of the thermally loaded parts of the engine (i.e. cylinder components) and components affecting the thermal load (air/exhaust gas system).

However, they are now working with condition monitoring of rotating machinery by vibration measurements and analyses, and the development of a condition monitoring or diagnosing system where all relevant components or systems were involved was their final goal.

As only a few of the faults had occurred on the Predikt II, installation had to be diagnosed and corrected by specialized engineers. However, this was mainly due to the highly skilled crew on this ship. But no special thought had been given to reliability of system hardware since this was a purely scientific installation.

For a commercial condition monitoring system special attention would be paid to system lay-out (easily exchangeable modules) man/machine communication and system "self diagnostic" ability. This was also verified in the paper of Mr S Espestøyl of Norcontrol A/S, presented at the IMarE 20 October 1977.

Replying to the question about the possibility of changing defect sensors without affecting the running of the engine: for some sensors (for instance piston ring proximity transducers), it was necessary to stop the engine before they could be replaced. However, no kind of engine reassembling was necessary.

Answering Mr Windridge: the vector elements used in the diagnostic system were the real deviation in condition parameter, measured by the standard deviation (measuring accuracy) of this particular parameter when the engine was new. All parameter values were assumed to be normal (gaussian) distributed. By this method parameters normalized by the common random variation of the parameter were obtained. Further, the principal of the diagnostic system was that the parameters did not vary at the same rate, but the different patterns of condition parameters indicated different failures under development in the engine plant.

Regarding Mr Mann's remarks the authors agreed with most of his points. The man/machine communication was perhaps the most important part of a condition monitoring system. In their experience a system which was difficult, or time-consuming to operate, would not be used even if the system gave valuable information.

The amount of information to be given should also be taken into careful consideration. They thought that a condition monitoring system might not be a success if the engineer had to search through a number of parameters to find the one he was asking for.

A system to be implemented on existing tonnage should comprise fewer measuring points than a system on a new ship because of the installation costs. The authors were sure that a system for condition monitoring of combustion and air-exhaust system would also be a good investment on existing tonnage.

The Working Party of Maintenance Engineering set out by the Minister of Technology in Great Britain in 1967 came to a different conclusion from Mr Victory concerning "satisfactory running of marine diesel engines for over 50 years". There is no doubt about the potential of improvement on maintenance engineering. Although the systems described were computer-based, the principal methods and knowledge could be used on an engine plant with a conventional instrumentation, i.e. the simple and very cheap Predikt I system.

Norwegian shipowners had used a system similar to Predikt II for years and the chief engineers had not reacted in the way Mr Victory had assumed, but they were very satisfied with the additional available information.

They agreed with Mr Victory's remark that "similar engines could not be expected to exhibit similar characteristics, wear rates or failure patterns. In the authors' opinion, that was the main reason for using on-condition maintenance systems.

Surrounding condition data were input data to the condition monitoring system, together with all other engine variables (temperatures, pressures, levels etc).

Mr Makinson had raised some questions about the pressure transducers used. In Demos a newly developed piezo-electric transducer was tested with promising results. That type of transducer did not need any forced cooling. The price was rather high (about £500 - including amplifier). Such transducers could be used both for combustion pressure and injection pressure measurements.

Dr Collacott had stated that the monitoring and diagnosing techniques might very well be applied to other system components than those dealt with in the paper. They fully agreed with Dr Collacott, and the use of vibration measurements and analysis in the condition monitoring of rotating machinery was now a project running at the Ship Research Institute of Norway.

A system had also been developed for the Norwegian Navy which applied condition monitoring and diagnosing techniques to the complete machinery installation. However, vibration monitoring was not a part of that system.

Mr Trousse had raised some interesting questions about the instrumentation system used in Demos. The parameters mentioned by Mr Trousse are obtained as follows:

Thermocouples (cromel/alumel) were used for both exhaust temperature and thermal load measurements. For exhaust temperature measurements the thermocouple was located in a pocket in the exhaust outlet pipe. The thermal load was measured as shown in Figure 5.

As shown in Figure 12 a digital thermometer was used for all temperature measurements. The accuracy of this measurement was $\pm$ 1% of F S (F S = 1000$^{o}$C). This accuracy was from the authors' point of view, quite satisfactory for this type of measurement.

MIP was obtained as follows: The cylinder pressure was recorded every ½ crank degree, i.e. 720 pressure recordings each revolution. In order to reduce the effects of process noise, the pressures were recorded over eight revolutions and the mean value calculated for each recording ($\bar{p}_i = \frac{1}{8}\sum_{i=1}^{8} p_i$). MIP was now calculated by means of the Trapezoidal Rule.

The measurement equipment was calibrated before being taken onboard. Maximum error 1% of F S (F S = 200 bar).

The accuracy of the mathematical models was hard to determine. The models used in their systems were empirical models, i.e. the accuracy of the models was at least no better than the accuracy of the data the models were based upon. Some of the models served their purpose quite satisfactorily, whilst there was room for improvements in others.

They were of the opinion that this new monitoring technique was a necessary supplement to the classical method of surveying the exhaust gas temperature. Previous work such as the Norwegian "Large Bore Project" and investigations made by Dr Ing E von Schnurbein and Dr Ing B Rau (presented at the Cimac Conference - 75) clearly showed this.

In reply to Mr Kirton, the condition monitoring systems described were only a part of the total results of many different projects at the Ship Research Institute of Norway during the last eight years. It was thereby quite difficult to give an exact estimate of the work attributed to the developing of those systems. A rough estimate should give approximately 20 man-years. Those projects had mainly been financially supported by the Royal Norwegian Council of Research, but shipowners as well as manufacturers had also contributed to that work.

Concerning the diagnostic system, the development of this had been fully financed by the Ship Research Institute of Norway during the last two years.

Whilst the proximity transducers in cylinder liners gave a most direct measure of the piston ring condition, the metal temperatures in the liner gave an earlier, but also more indirect and unreliable warning, on the piston ring condition. Enough knowledge had not yet been gained about the great fluctuation in liner temperature for this to be used as an alternative to proximity transducers. (Perhaps an accumulation of high metal temperatures could be a more reliable parameter on the piston ring condition).

Paper read at the Institute on 14 February 1978

# CARGO OIL PUMP INSTALLATIONS

Some Aspects of Design and Operation, and Problems Encountered

K.M.B. DONALD, B.SC., C.ENG., M.I.MECH.E., F.I.MAR.E.*


## Synopsis

The purpose of this paper is to put on record some aspects of design and operation of vertically and horizontally arranged cargo oil pump and associated machinery installations which were given consideration as a result of problems encountered in the early 1970's, for which Lloyd's Register of Shipping's Research and Technical Advisory Services were requested by Owners, Shipyards, and Manufacturers.

The contents are not a treatise on the specialised subject of Pumping and Piping, but are intended rather to reflect the range and types of problem dealt with in an advisory capacity and applying the principles of good engineering practice.


## INTRODUCTION

Liquid cargo pumps and the associated driving and transmission units installed in tankers play an essential role in trading throughout the world.

Crude oil carriers have always been equipped with their own pumping units almost solely for the purpose of discharging their cargo at oil terminals on completion of a voyage.

Products, chemicals, and L.N.G. tankers are equipped with cargo handling pumps to similar design, but this paper deals mainly with the crude oil carrier cargo pump installation, and some of the problems encountered by the Research and Technical Advisory Services Department.

It has been estimated (Ref. 1) that cargo pumps are operated at their maximum pumping capacity for an

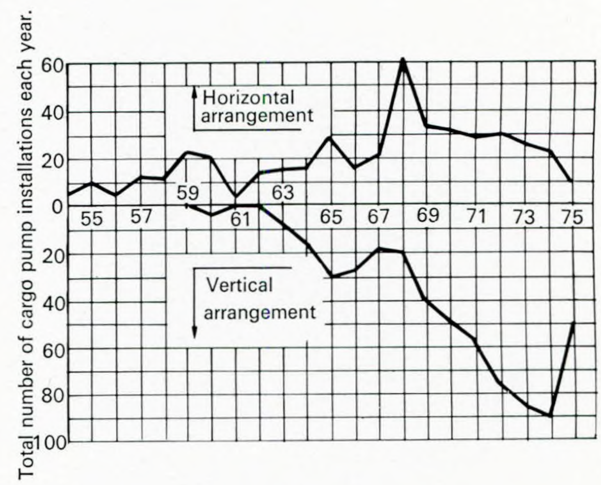



FIG. 1

Total number of cargo oil pump installations per year.

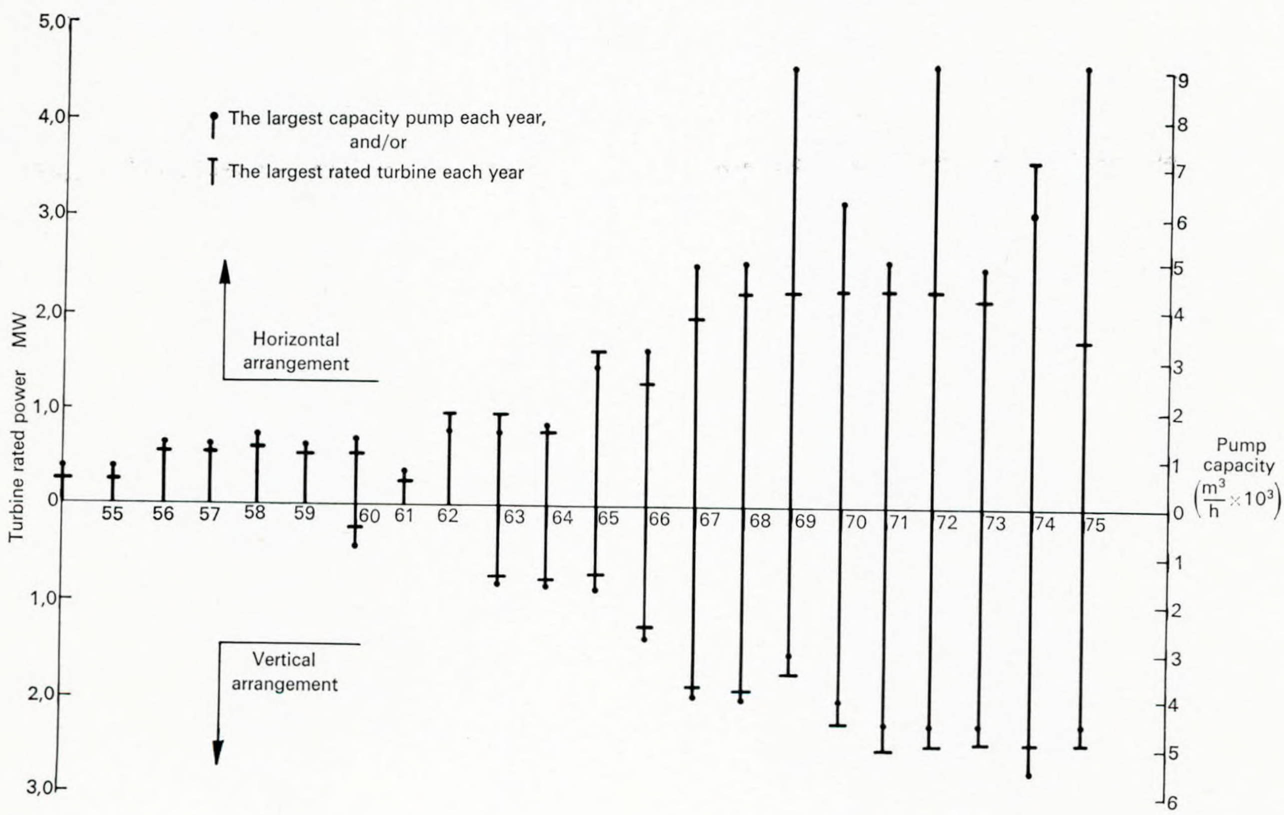



FIG. 2

Largest capacity pump, and/or highest rated turbine per year.

*Lloyd's Register of Shipping

average of about 1 hour per day or 40 hours total in a 40-day round trip, which makes it difficult perhaps to understand why expensive cargo space should be sacrificed to accommodate such machinery when a shore installation could be used on a more continuous basis. There are, however, considerable difficulties associated with any such proposal. Bottom or side discharge pipe connections, for example, would be particularly objectionable with regard to possible pollution hazards.

Although the sizes of crude oil carriers had been steadily increasing up to 1967, closure of the Suez Canal in 1967 provided the necessary impetus to accelerate the growth in tanker sizes, and thus lead to corresponding increases in the main engine ratings and cargo oil pumping capacity requirements of such vessels.

Accepting the on-board cargo oil pump (C.O.P.) installation as the clearly preferred practice, the question of whether the installation should be arranged vertically or horizontally remains open to debate. When the centrifugal pump was adopted as standard for cargo oil handling in the late 50's it was as a horizontal unit, but today most manufacturers are in a position to supply either type.

A study of the reference list prepared by Shinkokinzoka Industries Co. Ltd., Japan, has shown that their customers are showing a preference for horizontal units, and the company have gone on record as saying that they believe the world trend is reverting to horizontal installations. The same tendency is reported from the reference list of at least one other major pump manufacturer in Japan; Teikoku Pump.

From a reference list prepared by Jönköpings Mekaniska Werkstads A.B. (J.M.W.), the inference seems to contradict Shinko's forecast, for, as illustrated in Fig. 1, with the exception of a reversion to horizontal installations in 1967 and 1968, J.M.W's customers appeared progressively to favour the vertical cargo oil pump installation, a view shared by Ishikawajima-Harima Heavy Industries in Japan. As seen in Fig. 2, Shinko's second prediction that the horizontal installation is the preferred unit for capacities of 6000 m³/h and over is substantiated by the evidence of the J.M.W. reference list.

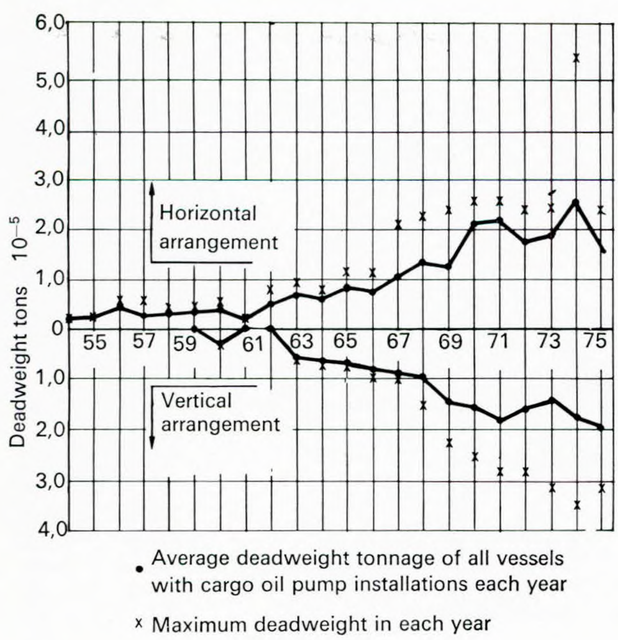



FIG. 3

The average, and maximum deadweight tonnage of tankers per year.

Finally, the inference in Fig. 3, which has also been prepared from the J.M.W. reference list, is that although the largest tankers afloat are equipped with horizontal cargo pumps, from about 1970 onwards vertical cargo pumps were the preferred installation for both V.L.C.C's and products carriers. (Products carriers of some 30 000 dwt reduce the yearly averaged deadweight figures, hence a greater difference between the average and maximum infers more such smaller tankers.)

## C.O.P. INSTALLATIONS ON CRUDE OIL CARRIERS

The C.O.P. installation, whether arranged vertically or horizontally, consists essentially of a single centrifugal pump, a transmission shaft which passes through a gas seal in either the deck or a bulkhead, and a driving unit. The drive for both steam turbine and motor tankers is usually through a single reduction gearbox, driven by a steam turbine, though there are instances (Ref. 2) of horizontal cargo pumps being driven through reduction gearing by high speed diesel engines, and the more recent innovation of gas turbine/electric drive for cargo pumps. (Ref. 3)

There are generally four cargo pumps installed on a large tanker, in addition to a ballast pump (which is often a smaller version of the cargo pump), and a stripping pump.

Generally, on a steam turbine propelled tanker, the steam to the C.O.P. turbines may be supplied directly from the main boiler at the full superheat condition, or from a steam/steam generator at lower temperature, and discharging to a separate atmospheric or sub-atmospheric condenser. The motor ship C.O.P. turbine will generally be supplied with steam from an auxiliary boiler at much lower pressure and temperature, discharging to either an atmospheric or sub-atmospheric condenser.

Typical steam condition for a 100 000 dwt motorship delivered in 1972 were inlet steam (absolute) 1·23 MPa, saturated, steam outlet (absolute) 0·03 MPa. By comparison, the steam conditions of a 280 000 dwt steamship of the same year were inlet steam (absolute) 5.9 MPa, 505°C, outlet (absolute) 0·13 MPa.

Steam driven C.O.P. machinery installations are by definition considered to be non-essential machinery and therefore Rules for the materials, design, construction, installation, or periodical Survey by Classification Societies, only apply in regard to the safety of the vessel. It would, in theory, be possible for a manufacturer to supply cargo pump machinery of sub-standard materials and workmanship, though of course the shipyard as the main contractor has the responsibility to ensure that the product chosen is of the appropriate standard to satisfy the owner's requirements. Further, a clean ballast pump, where fitted, is often a smaller but otherwise similar item of machinery supplied by the same manufacturer, and it is a Classed item and subject to inspection and Survey.

Only a few C.O.P. manufacturers make and supply the complete installation of pump, gearing and turbines. Some manufacturers specialize in pumps, others in gearing, and others again in steam turbines. The shipyard as the main contractor is responsible for engineering the total installation and for each of the component parts. It is not only the quality of the components which is important but, for satisfactory service, the overall installation. This concept is more readily accepted among manufacturers today with regard to main steam propulsion machinery where propeller, shafting, gearing and turbines are considered as a complete installation with regard to torsional vibration characteristics, and line shaft alignment and whirling, etc., but it is every bit as important with regard to C.O.P. installations also, especially when it is realized that C.O.P's are rapidly increasing in capacity and that the steam turbines which will power a single C.O.P. in the late 70's will have nearly the same power as the main propulsion machinery of the earlier oil tankers, namely 8000 s.h.p. (5·9 MW).

## SOME DESIGN ASPECTS OF VERTICAL CARGO OIL PUMP INSTALLATIONS

The accompanying diagram (Fig. 4) is a typical layout for the vertical C.O.P. installation, consisting of:—

(1) Pipework to and from the steam turbine casing.

(2) Stop valve and governor valve.

(3) Turbine casing.

(4) Turbine Curtis wheel. (Usually a two-row velocity compounded stage).

(5) Turbine rotor and gear pinion combined, with overspeed trip on bottom end.

(6) Gear casing and lubricating oil sump combined.

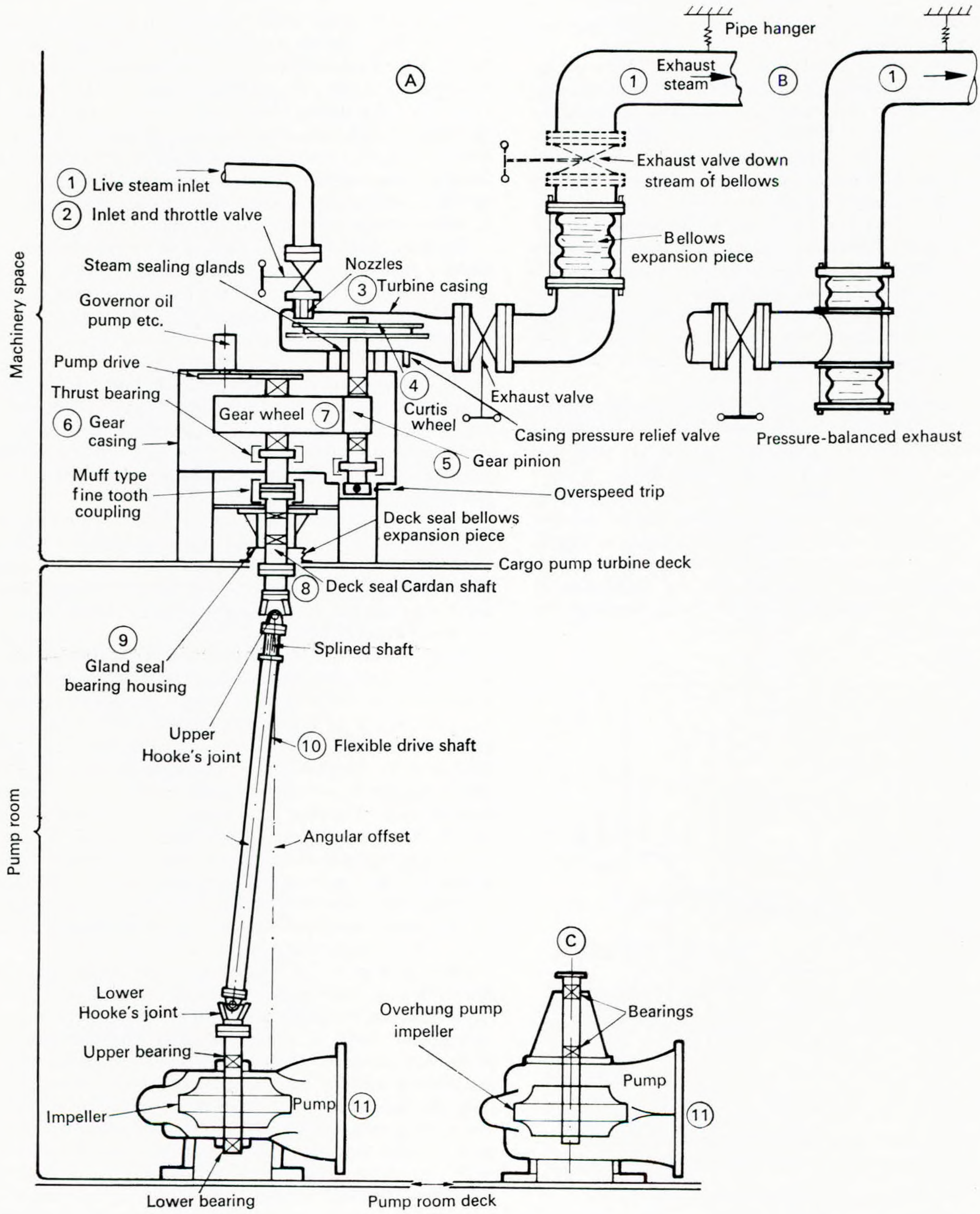



FIG. 4

Typical layout for the vertically arranged C.O.P. installation.

(7) Main gear wheel, with drives for lubricating oil pump governor oil pump, and speed governing mechanism.

(8) Deck seal Cardan shaft, having fine-tooth coupling onto gear wheel drive at top end, and solidly bolted to Hooke's joint coupling of drive shaft at bottom end.

(9) Deck seal and Cardan shaft bearing support.

(10) Drive shaft having two Hooke's joints, and axially splined floating joint at top end.

(11) The vertical cargo oil pump, usually a 'double' or balanced impeller, supported on bearings top and bottom or the recently introduced J.M.W. barrel design pump, having two top end bearings, but no bottom end bearing. (Overhung impeller.)

Examples of the machinery and pump installations supplied by Jönköpings Mekaniska Werkstads AB, Sweden, and Shinkokinzoku Industries & Co. Ltd., Japan, are illustrated in Figs 5 and 6 respectively.

Stal-Laval (Sweden) manufacture only the combined turbine and reduction gearing shown in Fig. 7, which can be arranged horizontally or vertically. Fig. 8 is a cut-away section of the turbine and gears.

Peter Brotherhood Ltd., Peterborough, England, also manufacture the combined turbine and gear installations as shown in Figs. 9 and 10.

Another manufacturer of steam turbines is Turbinenfabrik, J. Nadrowski, G.m.b.H., Bielefeld, West Germany.

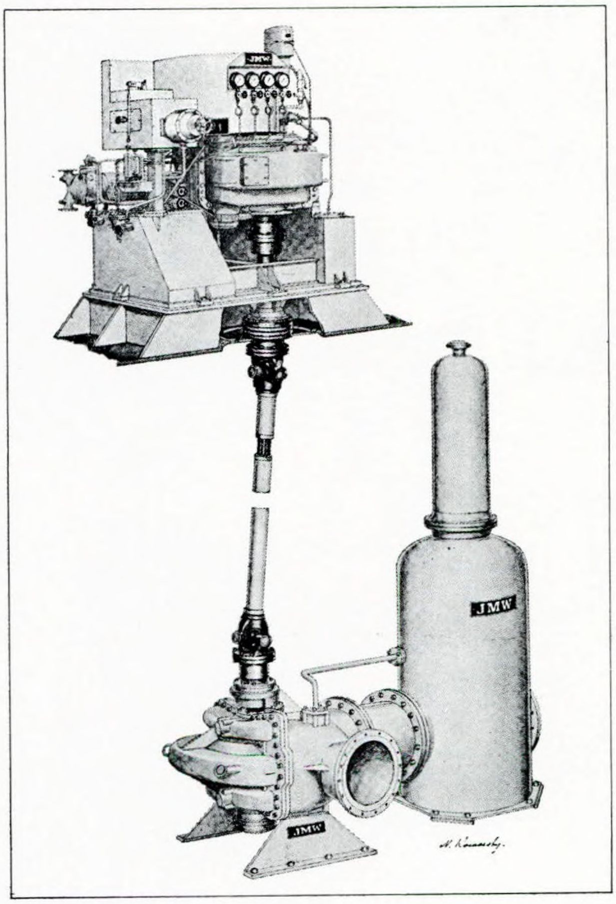
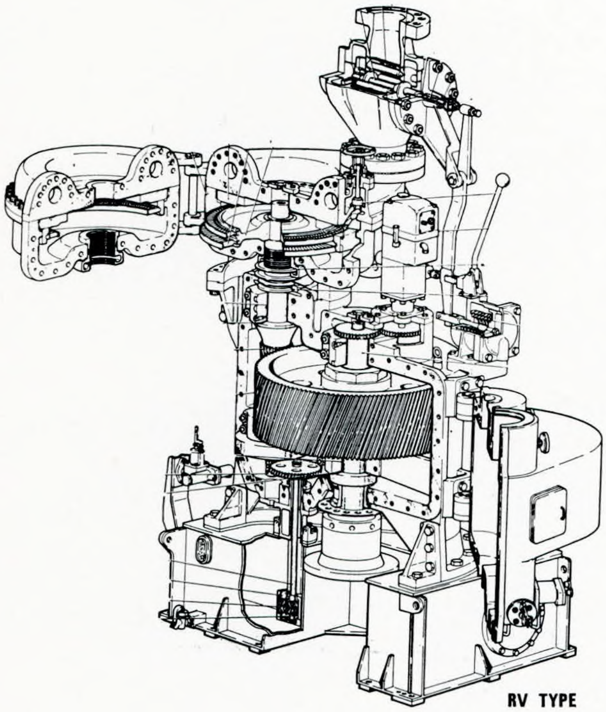



FIG. 5

C.O.P. and turbine machinery installation. (J.M.W.)



RV TYPE

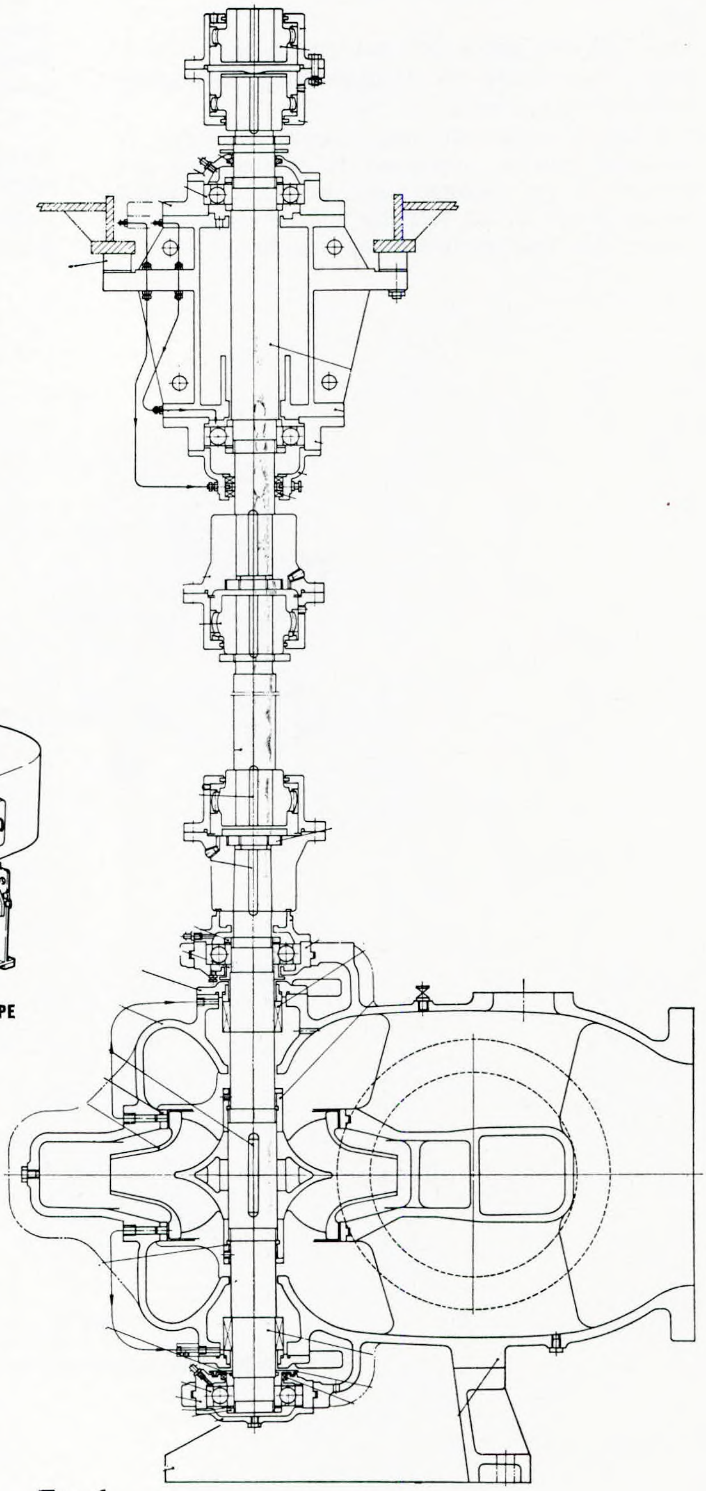


FIG. 6

Sectional view of a vertical C.O.P. installation. (Shinkokinzoku).

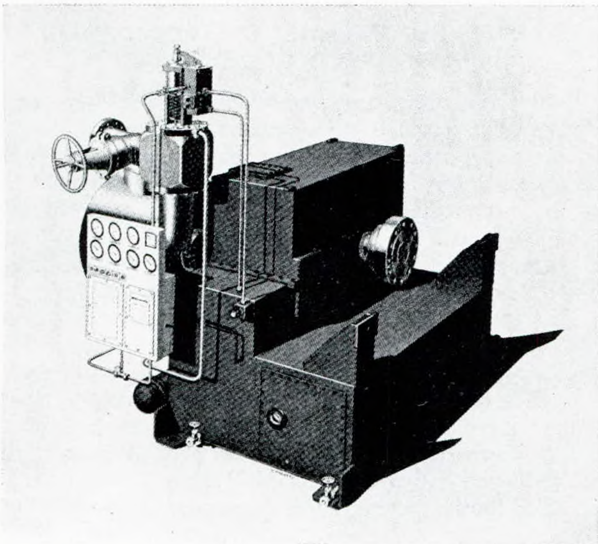


FIG. 7

'P.T. Series' combined turbine and reduction gear unit. (Stal. Laval).

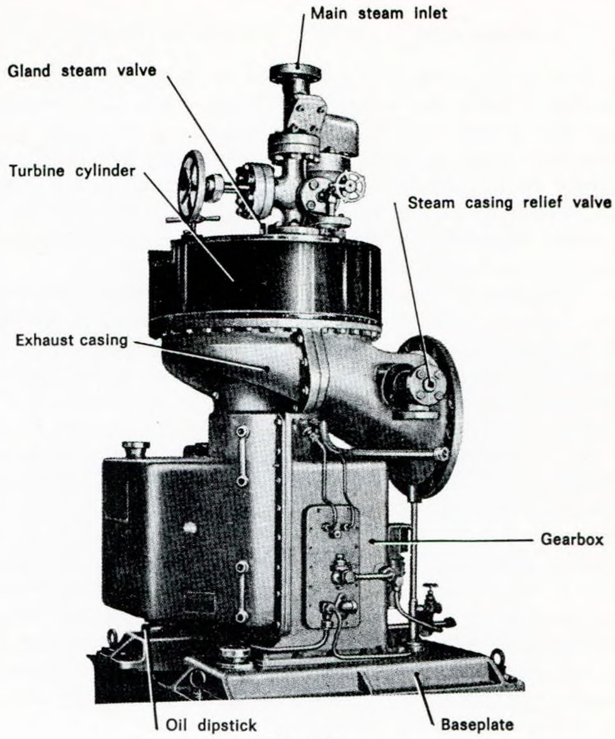



FIG. 9

Combined turbine and reduction gear unit. (Peter Brotherhood).

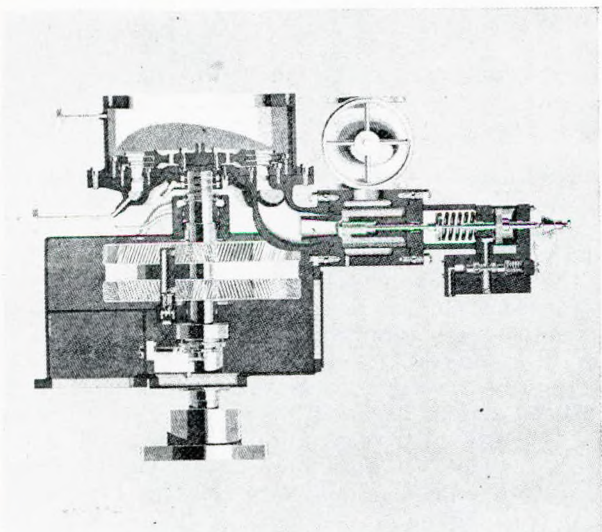


FIG. 8

Sectional view, 'P.T. Series', turbine and gearbox. (Stal Laval).

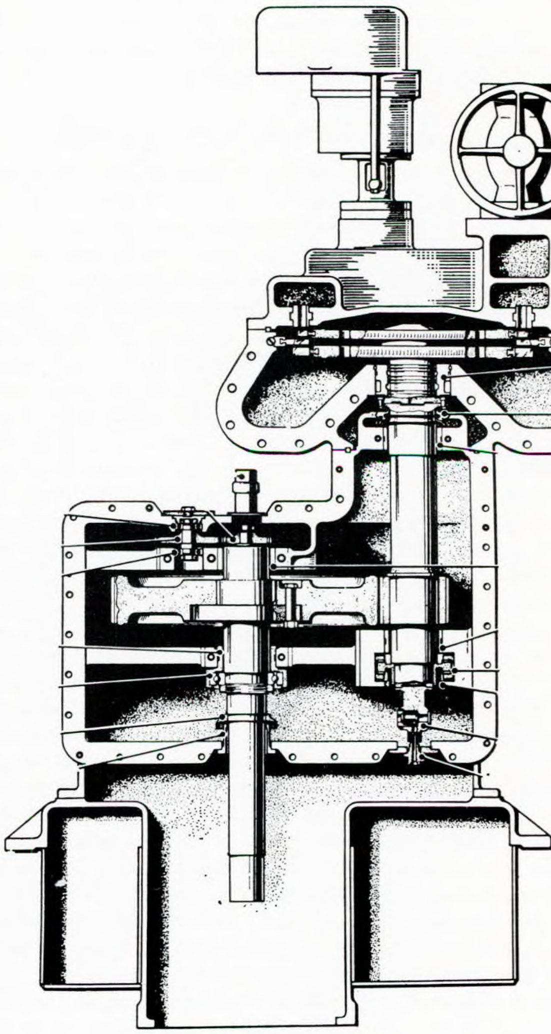


FIG. 10

Sectional view of turbine and gearbox. (Peter Brotherhood).

## CARGO OIL PUMP INSTALLATIONS

During the early 70's a number of Shipyards and Shipowners sought the assistance of the Technical Advisory Services Department of Lloyd's Register of Shipping in an advisory capacity, on problems associated with cargo oil pump installations in large crude oil carriers. In the course of investigations on vertical cargo pump installations, it became apparent that there were many more problem areas than had at first been evident.

Undoubtedly a number of the problems may have been the indirect result of the tanker building boom in the late 60's and early 70's, during which time the numbers and sizes of C.O.P's were rapidly increasing, when perhaps sometimes insufficient time and effort was concentrated in applying the principles of good engineering practice to not only the component parts, but also the overall installation. Further, as non-essential machinery, the Classification Societies had no direct involvement in the approval or Periodic Survey of these items of machinery unless specifically requested and paid for. Finally, by tradition, C.O.P. machinery could be operated by non-technical crew members, and the standard of routine maintenance would probably be minimal in many instances. The following are comments which highlight some of the problems dealt with and others which have been considered during the period of active consultancy.

It is very likely that the number of problems mentioned in this paper are but a small part of the total which occur worldwide, but which, for several reasons, may not be brought to the attention of the Classification Societies. The following therefore is in the nature of a review of the problems encountered for which the Society's Research and Technical Advisory Services have been requested in recent years, by Owners, Shipyards, and Manufacturers.

### I. LIVE AND EXHAUST STEAM PIPING

In the type of installation under consideration here and illustrated in Fig. 4(A), the turbine 'Curtis' wheel will be on the same shaft as the gear pinion, and carried in two bearings one on each side of the gear pinion, leaving the Curtis wheel and a length of shaft overhanging the bearing support. Such a turbine may be arranged vertically or horizontally. Thus the gearbox forms the main structural support for all rotating parts, and although the turbine casing is bolted to the top of the gear casing, its purpose apart from steam containment is to house the rotor shaft steam sealing glands and the fixed blades and nozzles. Steam inlet and exhaust pipes are also bolted to the casing, therefore the strength of the casing/gearbox attachment must be adequate to withstand any combination of pipe forces and couples, for any relative displacement between the gearbox and turbine casing could result in contact between fixed and moving blading (*see* Fig. 4(A)). There were instances where pipe forces, due chiefly to pressurization of the large diameter exhaust pipes, caused excessive moments to be applied to the turbine casing and seating bolts. In one instance it was found that bellows-type expansion pieces in the exhaust steam pipes were fitted upstream of the shut-off valve. The shut-off valve would be cracked open for warming through the turbine, but if the turbine stop valve should then be inadvertently opened up, prior to opening up the exhaust shut-off valve, the bellows could be pressurized. In addition to a possible bellows burst, the moment exerted on the turbine casing caused by the pressure differential acting on the projected area of the pipe could disturb the casing/rotor alignment. The casing relief valve should prevent serious pressurization, but the applied moment due to a pressure differential of one atmosphere in the pipe could in some instances be sufficient to deflect the casing, sufficient to reduce running clearances. A more satisfactory exhaust piping arrangement may be seen in Fig. 4(B).

There were also instances where insufficient attention was paid to correctly positioning water drains and dirt traps, and piping was installed sloping downwards away from drains.

### II. OVERSPEED, EMERGENCY STEAM SHUT OFF, AND SPEED GOVERNING

Turbine manufacturers usually specify a 10 per cent to 15 per cent overspeed as acceptable, but there have been instances where these limits have been exceeded, where, for example, a group of overload nozzles is provided, which may be detrimental not only for the high speed shaft but probably the slow speed shaft system as well.

If the turbine rotor critical whirling speed lies barely 10 per cent above the operating speed, then the limit of 10 per cent should not be exceeded, if, momentarily, possible heavy vibration is to be avoided in the turbine and gearing.

Where there is a speed governing valve and separate emergency stop valve (E.S.V.), as in the J.M.W. illustration (Fig.11), there is a possibility of boiler salts being deposited on the stem of the ESV, or of the packing glands being over-tightened, either of which will cause the valve to stick and delay the steam shut-off in emergencies.

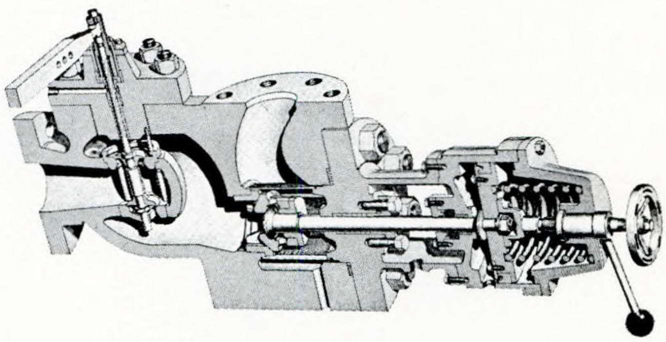


FIG. 11

Section through emergency stop valve and speed control valve. (J.M.W.).

Many manufacturers, it would appear, combine the duties of the ESV and governor valve in a single valve, which may be more reliable for both purposes, but could possibly lead to other problems.

The overspeed trip mechanism is generally located on the lower end of the turbine rotor/pinion shaft (i.e. below the pinion) as neatly illustrated in a cross-section through the J.M.W. rotor (Fig. 12), a design feature which has been perhaps unjustly criticized in recent years, for there is no other position on the shaft which would be acceptable for the conventional overspeed trip mechanism of the type which employs the displacement of a springloaded ring or slug to strike the finger of an oil relay trip linkage. There is no reason why a suitable electrical trip device should not be developed to operate at some position on the shaft above the pinion provided:—

(a) it could reliably withstand the more adverse environment,

(b) it added no mass to the shaft,

(c) it did not reduce the shaft stiffness at any position.

The speed governing mechanism, and the lubrication and relay oil pumps are all usually driven off the top or free end of the gear wheel shaft.

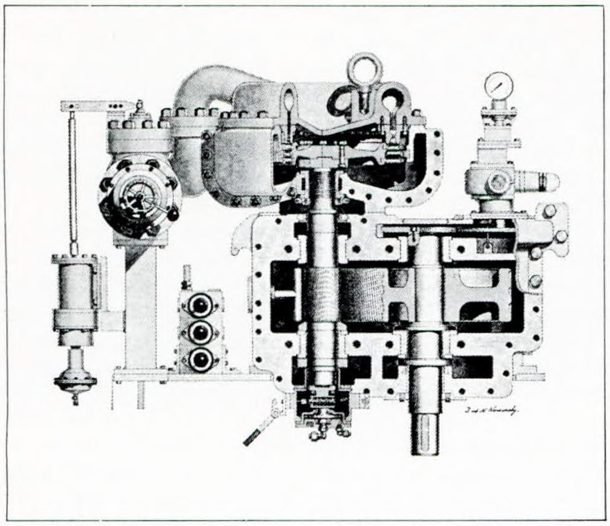


Fig. 12

Section through the turbine and gearbox. (J.M.W.).

## III. CRITICAL WHIRLING SPEEDS

### 1. High Speed Shaft

With regard to turbine rotor/gear pinion (high speed) shaft, there are a number of important aspects of design which should be mentioned here.

Firstly, since the cargo oil pump will be operated over a wide speed range, the high speed shaft should be designed to be under-critical, or sub-critical, which means that the first critical whirling speed should be well above the maximum operating speed (and the overspeed), for it must be assumed that even a vertically mounted shaft will vibrate at its critical speed unless the centroid of every section along the shaft lies on the axis of rotation, a state of perfect balance which cannot be achieved in practice. Thus when the designer has selected the most suitable pump speed and single reduction gear ratio, for the duty required, the turbine/pinion shaft speed will be known, and the best rotor/pinion dimensions can be considered to ensure that the rotor is sub-critical.

Without going into too much detail at this stage it should be understoood that the high speed shaft first critical whirling speed (whether rigidly mounted or allowing for oil film and bearing support flexibility), is dependent upon its dimensions and construction, or in physical terms, the rotor stiffness and mass distribution.

One of these parameters is the overhung mass, i.e. the total mass of the wheel and blades on the top end of the shaft. Other parameters are the length of the overhang, and the diameter of the overhanging shaft, and so on, each of which has a direct influence upon raising or lowering the critical speed.

For example, all other factors being equal, any increase in overhung mass will lower the critical speed. Similarly, increasing the overhang and reducing the overhung shaft diameter will each tend to lower the critical speed. If care is not taken at the design stage in assessing the effect of each of these various parameters, then the critical speed could coincide with the maximum operating speed and severe vibration could result, which can then only be avoided by reducing the operating speed, or by complete re-design of the rotor.

A distinction must also be made between the first critical speed calculated assuming rigid bearing support, and that which occurs in practice due to flexible bearing support, since the latter will always be lower.

### 2. Low Speed Shafting System

The low speed shafting comprises the reduction gearwheel, a deck seal Cardan shaft, the flexible drive shaft, and the pump impeller shaft. The longest single length of shaft for the vertical installation is the flexible drive shaft which may be from 3 to 4 metres in length having a Hooke's joint flexible coupling at each end as shown in Fig. 5, or the fine-tooth coupling shown in Fig. 6. The gear wheel shaft and pump impeller shafts will probably be offset from each other so that the flexible shaft with Hooke's joint will run with anything from $\frac{1}{2}°$ to 5°, or more, angular displacement at each end. (The barrelled fine-tooth coupling shown in Fig. 6 should not exceed an angular displacement of $\pm 15$ minutes according to Shinko, but they will supply Hooke's joint couplings, Falk couplings, or diaphragm couplings.)

Running speeds for the slow speed shaft can be up to about 1800 r.p.m., being geared down from the turbine high speed shaft in a range of from 4:1 to 8:1, depending on the manufacturer's design.

Calculation of the slow speed shafting critical speeds should include allowances for the flexible couplings, bearing flexibility, and possibly a lumped non-rotating mass to simulate the deck seal bearing housing carrying the Cardan shaft (as seen in Fig. 13).

Such allowances are difficult to estimate with any real certainty since, for example, the deck seal bearing housing flexibility, and lumped mass effect may be dependent upon the engine seating and deck flexibility, etc. Generally, it is advisable to compare calculated results with those measured during tests after installation, and, as for the high speed

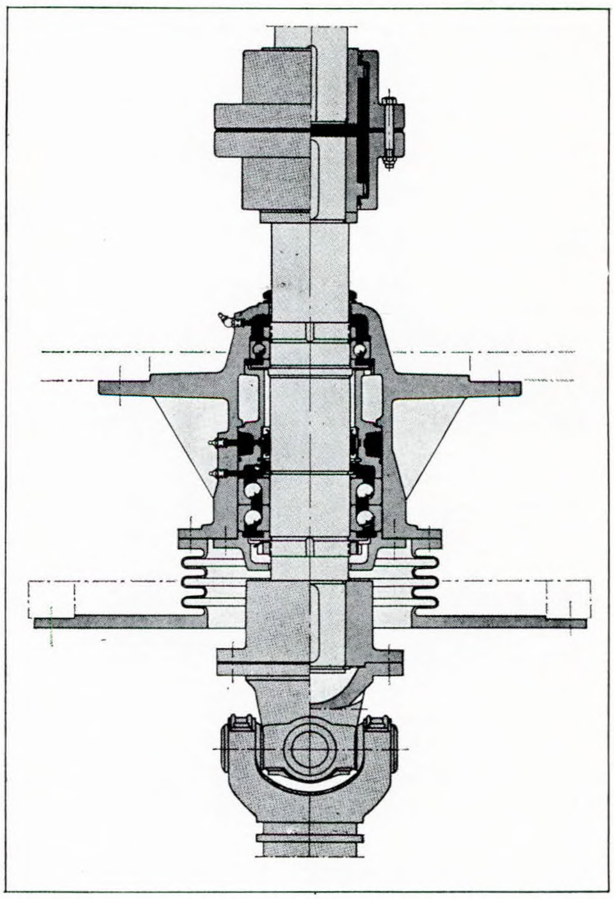


Fig. 13

Section through Cardan shaft bearing housing and deckhead gas seal. (J.M.W.).

shafting, the low speed shaft system should be designed for sub-critical operation.

In addition to the first order (once per revolution), first and second mode critical whirling speeds arising from residual unbalance, there may be a further potential source of excitation associated with the Hooke's joints at each end of the flexible shaft.

Such joints produce a small second order oscillation in angular motion at each joint, presumably accompanied by second order radial force effects of opposite sign, which could excite the shafting system at its lowest natural frequency at a rotational speed below the maximum operating speed. This aspect has not been fully explored to date and it will probably depend upon the support flexibility at each end, length of Cardan shaft, etc., but further examination would require measurements *in situ*.

## IV. TORSIONAL VIBRATION

The torsional vibration characteristics of the complete shafting system should be calculated, and the natural torsional frequencies compared with the possible sources of excitation such as gear pitch error, second order Hooke's joint torsional excitation, pump impeller vane impulse, etc.

The torsional excitation due to a Hooke's joint at each end of a flexible shaft should be self-cancelling provided that the angular displacement at each joint is identical, but this may not be the case if, for example, the pump impeller axis and deck seal Cardan shaft axis are not aligned to be parallel when installed with an offset.

Unfortunately, there is insufficient information available at present to assess the magnitude of the various forms of torsional excitation, or the damping characteristics. However, it could be argued that even with the limited data available calculation of the torsional frequencies would be a worthwhile exercise. Case 2 (Section XI) provides an example.

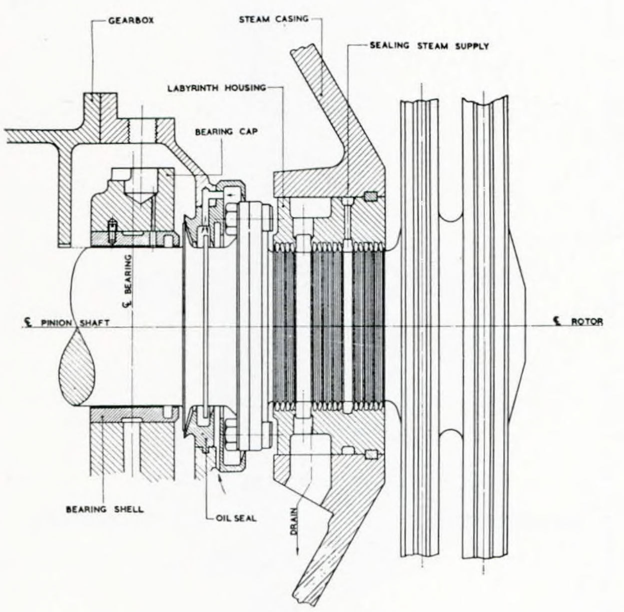



FIG. 14

Arrangement for bolting turbine wheel to gear pinion shaft. (Peter Brotherhood).

## V. CURTIS WHEEL ATTACHMENT

All manufacturers of the combined pinion and overhung wheel design of rotor attach the Curtis wheel separately, but there are many variations of the two basic methods of attachment, namely a continuous bolted-on disc (no central hole), or a disc with a central hole keyed to the shaft.

Stal-Laval (Fig. 8) and J.M.W. (Fig. 12) favour bolting the wheel to a spiggoted flange on the end of the shaft within the turbine casing, whereas the Peter Brotherhood design (Fig. 14), takes the flange outside outside the steam space and nearer to the upper bearing. Shinko (Fig. 6) and the Nadrowski design prefer the wheel with central hole keyed to the shaft and held with a locking nut.

The bolted-on wheel in the steam space may result in a marginally greater overhung mass, but in all cases the security of the wheel depends upon the locking nuts. Shinko allow the end of the shaft to protrude, and carry this through a recess in the top half of the casing. It is not a bearing, but could perform as a steady bearing in cases of extreme vibration, or as a fail-safe shaft end retaining bearing.

Nadrowski have reduced the overhung mass of the wheel by electro-chemically machining out the two rows of blades from the solid wheel rim. This has the advantage over the conventional method of fitting separate blades in circumferential grooves in the wheel, of reducing the wheel rim depth and width, which in turn reduces the required thickness of the wheel and hub. There are disadvantages, however, for no shrouding is fitted, which results in windage and tip spillage losses, and from the blade vibration aspect there will be no interface friction damping in the roots or shrouding. These are disadvantages which are quite distinct from the problems arising from electro-chemical machining of blading from the wheel rim, though the technique may be improved in time.

## VI. TURBINE CASING MATERIAL

The steam chest containing the steam inlet flange, the inlet steam chamber and nozzles are made from cast steel to withstand the high pressure and temperature environment, but after expansion through the nozzles, the lower pressure and temperature could be accommodated by a cast iron casing. However, if for any reason the inlet valve were fully open, and an exhaust valve shut, then theoretically, if the relief valve did not operate, the casing could become fully pressurized, and may not be able to withstand the full steam pressure.

There is a further possible hazard however, one which should always be carefully considered, and that is the physical containment of moving blades which break off, a wheel which detaches from the shaft, or a burst wheel due to excessive overspeed. If there is any doubt about containment then preferably the casing should be made of cast steel having the required strength and design to absorb completely the energy of the wheel and contain the wheel without fracture, under any of the contingencies mentioned above.

## VII. TURBINE BLADES, TENONS AND SHROUDING

There have been a number of instances of blade failures due to metal fatigue. Such failures have been either in the root, or the tenon of the blade. Loss of a blade, or section of shrouding could result in severe rotor unbalance which could consequently damage bearings, fixed blades, gear teeth, etc., if the installation is operating at or near the maximum speed when the blades fail.

There are probably some instances of blade failure from one cause or another, for which the Society's Technical Advisory Services have not been requested, but of those investigated on behalf of owners or manufacturers, failures were generally confined to the second row of blades of the two-row velocity compounded stage, and the majority suffered tenon fatigue, though there have also been blade root failures in some instances.

The first row blades for such two-row C.O.P. turbines are generally short and stubby (low aspect ratio), having roots which are often longer than the aerofoil section. This makes natural frequency prediction somewhat hazardous if the point of root fixity at various operating speeds is not known. The sketch in Fig. 15 illustrates the typical proportions such a blade could possess. After the blades are assembled in the wheel, a wedged closing piece tightens the abutment of blades circumferentially. When the wheel is rotating at high speed the centrifugal force of the blades acting on the root lands tends to force open the wheel rim, but should be prevented from doing so by the side lugs on the blade root. In this way an effective 'clamping' should result between wheel rim and blade. The effective 'free length' of blade will then include the aerofoil length and part of the root to the side lugs (A). If at lower rotating speeds the rim is not opened out sufficiently to effectively clamp the root side lugs then the effective blade length is extended down into the root lands, to (B). In the proportions shown in the sketch, such a change of effective length would reduce the lowest natural frequency of vibration by about 50 per cent, and reduce the higher mode frequencies by similar amounts. Natural frequencies are in the Kilo Hertz region. A further complication is introduced if any of the points of 'fixity' of the root are considered as having a finite stiffness rather than being treated as rigid.

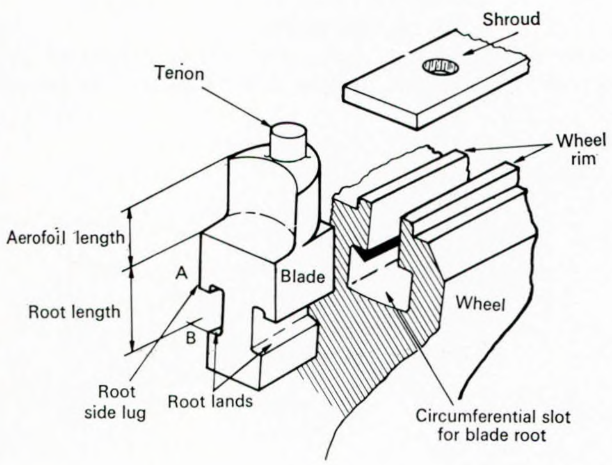



FIG. 15

The first row blade in a 2-row 'Curtis' wheel.

Finally, the interaction between wheel modes and packet modes can sometimes be significant depending upon their relative stiffness and mass distribution, etc.

Steam is admitted to the first row blades through groups of nozzles which occupy an arc of two or possibly three quadrants known as 'partial admission'. The significance of which, in relation to the cyclic loading imposed upon the moving blades, is far more severe than the 'full admission' (complete 360° annulus of nozzles), common in multi-stage turbines, since each blade passing into and out of a small arc of nozzles experiences a cyclic variation from zero load to full steam load to zero load again.

Second row blades are usually longer and thinner in sections but the same remarks apply with regard to calculation of natural frequencies on a somewhat reduced scale. The steam admitted to the second row blades passes through an arc of fixed blades attached to the turbine casing but because steam velocities are lower at inlet to the second row blades, the cyclic loading is also lower.

The greater incidence of second row blade failures may seem to contradict the foregoing, but the more robust first row blades are designed for operation at relatively conservative levels of stress because of the more arduous, and less easily predicted conditions prevailing at outlet from the nozzles. Furthermore, the lowest natural frequencies of the first row blades are, almost invariably, higher than the nozzle passing frequency, whereas the longer second row blades of smaller section have lower natural frequencies, so fixed blade passing frequency could coincide with the first tangential out-of-phase modes of packet vibration (the so-called clamped-pinned modes), at or near the maximum speed. This may well explain failure of second row blades at the root, but tenon fatigue failures can be atttributed in most instances to a combination of excitation of a blade natural frequency with a higher mean stress in the tenon. From the sketch in Fig. 16 which illustrates the relative blade sections which may be employed it will be seen that

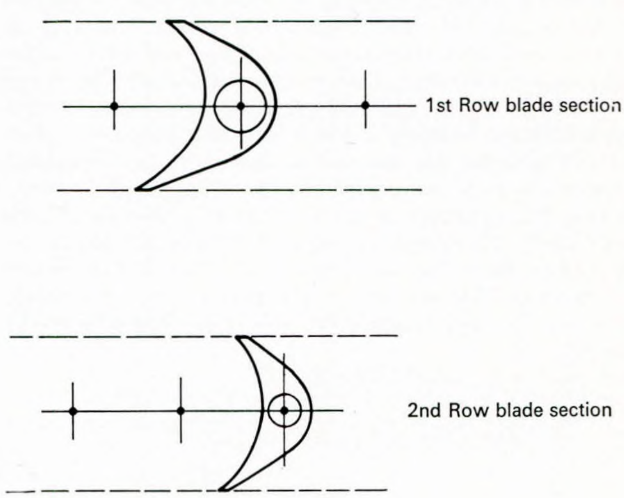



FIG. 16

The relative size of blade sections in a 2-row 'Curtis' wheel.

the size of the tenon which can be accommodated on each blade section, allowing for a suitable footing at the base of the tenon is dependent upon the blade section. Thus even if an allowance is made for a slightly reduced shroud thickness and pitch (and hence centrifugal force in the second row tenon), the areas of the tenons are in the ratio of 2:1, and the centrifugal stress in the second row tenons will thus be greater than those in the first row by about the same ratio.

With regard to hand-fitting of the shrouding, the pitch of the holes to be drilled in the shroud should be marked off to fit each 'as-fitted' batch of blading. However, minor errors may necessitate filing the tenons if the pitch, or the axial alignment of the blades does not correspond to that of the shroud band holes. In doing so, notches may be formed in the tenon roots, but additionally, the same amount of metal removed from each of the two sizes of tenons will reduce the effective load carrying area of the smaller tenons by a factor of two compared with the first row tenons. Tenon head formation may also be important, for if the tenon heads are formed by means of a slow hydraulic squeeze technique instead of the time-honoured hammer-blow peening method, the results will be quite different. The rate of plastic deformation is different in each case. The slow squeeze may produce a 'fold' of metal at the base of the tenon, introducing an effective notch, whereas the hammer-blow method mushrooms only the tenon head. Shroud holes must be correctly chamfered, tenon bases properly radiused, and so on.

## VIII. TURBINE ROTOR AND GEAR PINION COMBINED

A number of manufacturers had adopted the shrunk-on gear pinion construction some of which suffered cracking and shaft failures from rotating bending fatigue. These cracks originated from fretting on the shaft surface at entry to the shrink fit, in way of the shaft shoulder fillet. There could be many valid reasons for choosing to build-up the rotor by shrink fitting gear pinions, thrust collars, and the like, and provided that proper attention is paid to the details of design, manufacture, and shrink fitting, such a construction may be satisfactory in some instances.

However, with regard to the turbine/gear pinion shaft critical whirling speed, the shaft stiffness and mass distribution between the bearings are also important parameters. A reduction in stiffness between the bearings will reduce the critical speed. (All other factors remaining constant.) A built-up shaft with shrunk-on gear pinion and thrust collar will in most instances exhibit a lower critical whirling speed compared to a dimensionally identical solid shaft, because the transverse bending stiffness will probably be slightly reduced whereas the distributed mass will be unchanged. Perhaps the most compelling argument against the shrunk-on gear pinion design is the consideration that should the rotor shaft fail in way of the gear pinion, the wheel and top end of the rotor could run away, since the overspeed trip is on the bottom end of the gear pinion/rotor shaft, therefore the inlet steam valve would not immediately be shut off.

## IX. GEARING

Most manufacturers employ a single reduction spur or single helical gear which makes it necessary to fit thrust bearings on both the high speed and low speed shafts. The rotor/gear pinion shaft and the main gear wheel shaft bearings may be a conventional shaft collar and thrust pads located below the bottom journal bearing, or in some designs the journal bearing sides act as thrust surfaces.

It will be seen from Fig. 8 that the Stal-Laval C.O.P.T. reduction gears are the double/helical type, consequently, the turbine rotor/gear pinion shaft has no thrust bearings, the axial location of both the main wheel and pinion/rotor rotating elements being maintained by the main wheel thrust bearing.

## X. LOW SPEED SHAFTING SYSTEM

Referring to the sketch in Fig. 4 of the vertical C.O.P. installation it will be seen that the low speed shafting system comprises four separate shaft elements, viz:—

(Vertical Arrangement)

1. The gear wheel and shaft
2. The deck seal stub shaft
3. The flexible drive shaft
4. The pump impeller and shaft

Whereas in the case of the horizontal C.O.P. installation (Fig. 17) there are only three basic low speed shafts, viz.:—

(Horizontal Arrangement)

5. The gear wheel shaft
6. Cardan shaft with fine tooth couplings
7. Pump impeller shaft.

The deck seal stub shaft (Fig. 13) is supported in a separate housing and runs in ball bearings. The housing

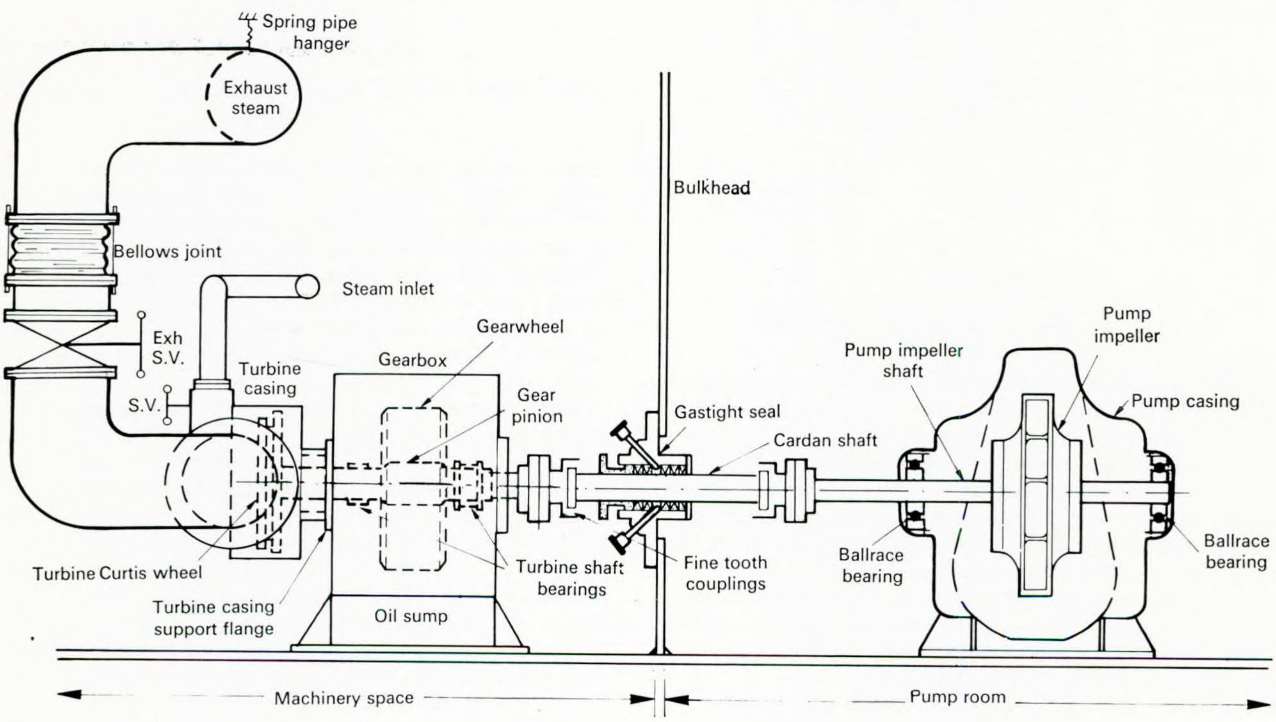



Fig. 17

Typical layout for the horizontally arranged C.O.P. installation.

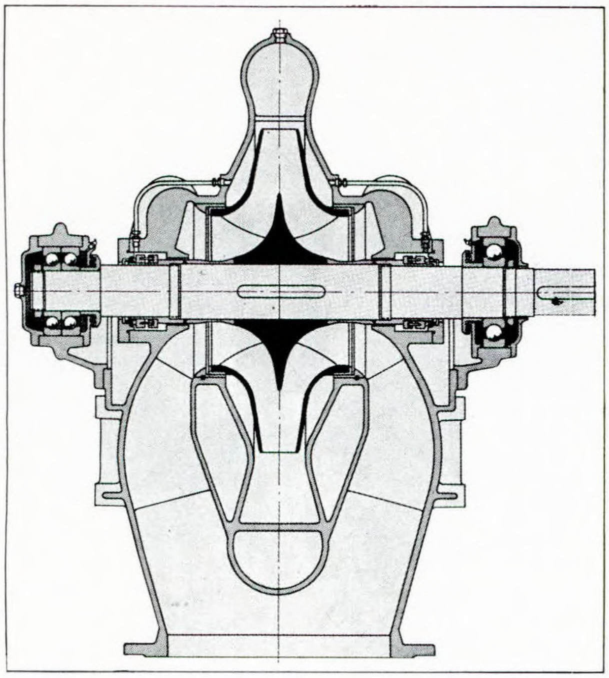


FIG. 18

The double volute cargo oil pump. (J.M.W.).

is flange mounted onto the lower part of the engine seating immediately below the gearbox, and a stainless steel bellows piece is fitted between the ball bearing housing and the deck to provide the gas seal. The stub shaft is flexibly coupled to the gear wheel output shaft through a fine-tooth muff coupling, while the lower end of the stub shaft is solidly coupled to the flexible drive shaft. An alternative type of gas-tight deck seal can be seen in Fig. 6.

The flexible drive shaft is of the 'Hardy-Spicer' type (Fig. 5), having Hooke's joints at each end, and the shaft is in two parts to permit axial float through male and female splines. (The alternative fine-tooth coupling transmission shaft is illustrated in Fig. 6.)

Finally, the impeller shaft is mounted on ball bearings at the top and bottom ends of the pump (Fig. 18) and solidly coupled to the lower Hooke's joint of the flexible

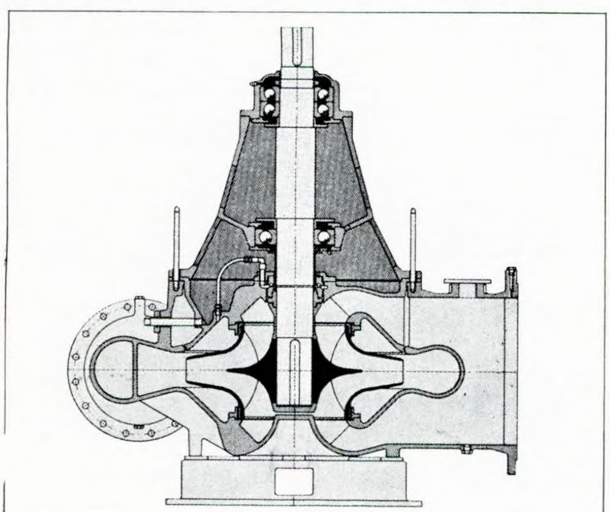


FIG. 19

Type Z233 radially split pump casing and overhung impeller. (J.M.W.).

drive shaft. (Alternatively, the overhung impeller type pump shaft may be supported in two bearings above the impeller, as illustrated in Fig. 19.)

In the horizontal arrangement the Cardan shaft passes through a gas-tight gland seal located in the bulkhead be tween engine room and pump room, one arrangement of which is shown in Fig. 20.

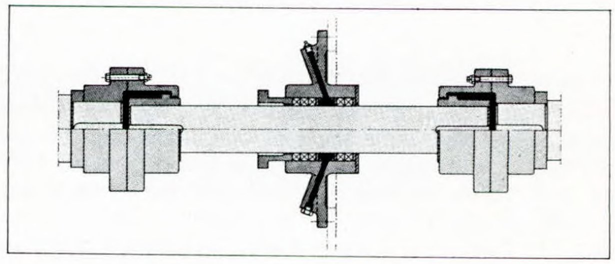


FIG. 20

Bulkhead gas-tight gland seal for the horizontally arranged C.O.P. installation. (J.M.W.).

2. **The deck seal and stub shaft**

The bearing housing (Fig. 13) is effectively cantilevered from the engine seating, so that in addition to ensuring adequate transverse and vertical stiffness for the stub shaft bearing housing and seating as mentioned in Section III, detailed attention should be given to the alignment and lubrication of the fine-tooth muff coupling with sufficient access for alignment and subsequent maintenance. Further, should the gear wheel thrust bearing fail, the stub shaft thrust ring may have to support the gear wheel weight for a period of time, depending upon axial clearances of these components.

3. **The flexible drive shaft**

There are known to have been some failures of this type of shaft manufactured by Gelenkwellenbau Gmbh, Essen, in which peripheral technical investigation was undertaken on behalf of owners and shipyards, but to date the overall conclusions and effects of certain remedial measures which were to be implemented are not known to the Author.

Problem areas appeared to be associated with the balancing procedures, Hooke's joint bearing failures, and possibly initial assembly, newbuilding installation, and maintenance.

The crucifix of the Hooke's joint is mounted in four roller bearings, the outer race of each being held in split housings. Each bearing keep is secured by two counter-sunk socket head bolts. In one case examined, the bolt heads had fractured due to fatigue, but no definite conclusions as to the cause of fatigue was known at that time. Re-design of the bolts is believed to have been undertaken. In another case examined, no such cracks were found, in the bolt head but some doubt was expressed as to the seating of the heads on the spot-faced keep.

One instance has been quoted of tack welding small pieces of steel to the middle of the hollow Cardan shaft for balance purposes. The shaft subsequently failed by fatigue initiated in way of the welds which had not been stress relieved. Another instance has been quoted of balance being achieved by controlled bending of the Cardan shaft.

The manufacturers issue specific instructions for routine maintenance of the flexible drive shaft, and for the transport and lifting of complete shaft assemblies, yet the existence of these instructions may not always be brought to the notice of the owner or the shipyard.

Although it was generally agreed that the axis of the pump and engine drive shaft should be aligned for parallel offset, the degree of angular displacement thus applied to each joint was open to question. The manufacturers were reported to have stipulated a maximum of 15° angular obliquity but this was considered excessive and values of 3° and 1° were discussed, but no definite conclusions were reached pending measurements *in situ,* particularly to establish the order of magnitude of any changes of alignment due to hull deflections, local distortions, etc., and to determine the influence of the axially splined expansion joint in the shaft.

4. **The pump impeller and shafting**

Two main problems associated with cargo oil pumps have been reported, these being erosion of the impeller due to cavitation, and failure of the pump bearings. It has been suggested that the bottom end bearing of the conventional, vertically arranged pump is not readily accessible for routine lubrication and maintenance and may tend to be neglected, however, it is not known if the incidence of failures supports this supposition.

## XI. SOME CASE HISTORIES

The following are brief descriptions of two cases investigated when acting in an advisory capacity for Owners, Shipbuilders, or Manufacturers on various occasions.

**Case 1. Products Carrier. 30 000 dwt. Delivered 1972. LR Class**

During commissioning tests at the shipyard on the four steam turbine driven cargo oil pumps, severe turbine vibration was reported when the turbines were operating on load between 5600 r.p.m. and 6000 r.p.m. (Service speed 6264 r.p.m., maximum overload speed 7200 r.p.m.)

The turbine manufacturers stated that the first critical whirling speed of the rotor on rigid bearings had been calculated to lie between 7796 r.p.m. and 8300 r.p.m., well above the maximum overload speed of 7200 r.p.m.

Previous vibration had been attributed to loss of turbine blades and, having since strengthened the blades, none had failed. The fitting of anti-whirl bearings had not been successful in limiting vibration, and the pumps could not be tested. The Yard requested the Society's assistance.

Briefly, a rough calculation had indicated that the critical whirling speed on rigid bearings was probably nearer 6500 r.p.m., therefore a detailed computer-aided analysis was undertaken using the Society's shaft whirling programme, LR 272, for both the rigid bearing case, and for a number of different 'composite' bearing flexibilities.

**Original Rotor. Analysis (a)**

Curve (a) in Fig. 21, represents the calculated relationship between critical whirling speed (r.p.m.), and bearing flexibility, for the original as-built rotor, as shown diagrammatically in Fig. 22(a).

The object was to raise the critical speed so that the rotor would be sub-critical, i.e. the critical whirling speed would lie outside the maximum overload operating speed of 7200 r.p.m., but the number of options available was limited unless a complete turbine re-design was to be undertaken.

**First Modification. Analysis (b)**

The first modification (Fig. 22(b)) was to increase shaft diameters, and to change the original shrunk-on pinion to one integral with the shaft but retaining the separate thrust collar. Only a limited increase in diameter of the overhung shaft end was possible since the steam sealing gland housing could not be changed. The first modification raised the rigid bearing whirling speed by some 3000 r.p.m., curve (b) in Fig. 21, which appeared to be a satisfactory solution to the problem.

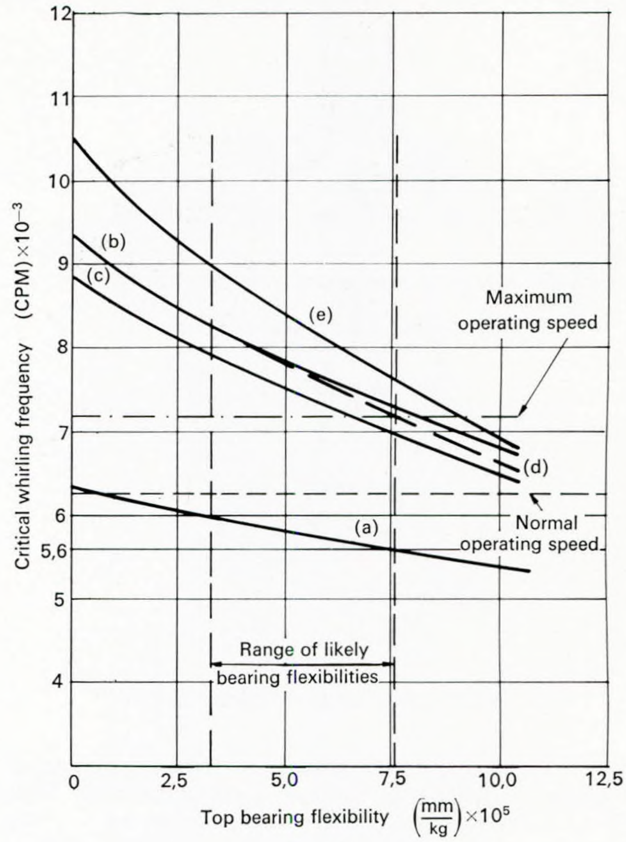



FIG. 21

Relation between the turbine/gear pinion shaft critical whirling speed and bearing flexibility.

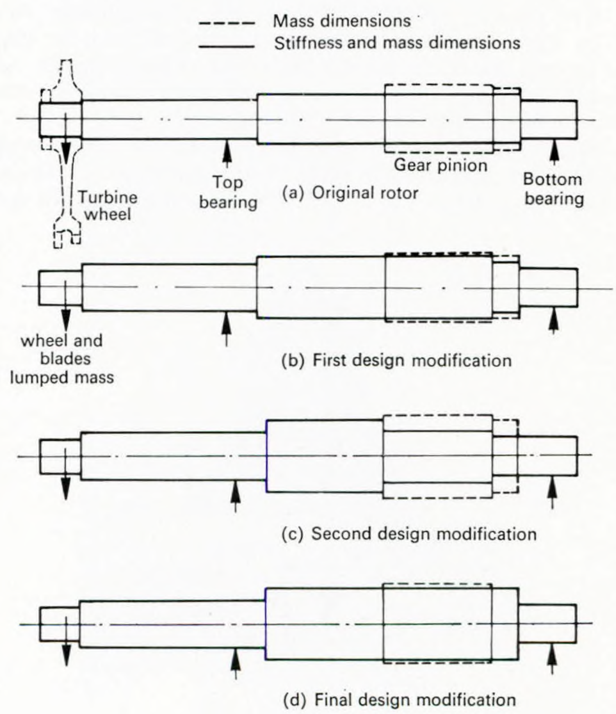



FIG. 22

Representation of the shaft dimension modifications chosen for analysis in Fig. 21.

**Second Modification. Analysis (c)**

The manufacturers agreed to the proposal to increase diameters but wanted to retain the separate pinion and thrust collar. They also wanted to reduce the top bearing length which effectively reduced the bearing span (and increased the overhang) by 20 mm.

In view of these somewhat adverse requirements it was agreed that the shaft diameter between the bearings, and the pinion diameter be increased by 20 mm, as shown in Fig. 22(c).

Curve (c) in Fig. 21, illustrates that the rigid bearing critical speed calculated for the first modification was reduced by about 500 r.p.m.

**Third Modification. Analysis (d)**

The manufacturers were advised about the inherent dangers of fitting a separate pinion and collar, and accordingly the proposals put forward in analysis (c), were modified as illustrated in Fig. 22(d), assuming the pinion and thrust collar to be integral with the rotor. Curve (d) in Fig. 21, shows that the increase in critical speed was achieved for the rigid bearing case, but the benefit diminished with increased bearing flexibility.

**Final Analysis. (e)**

The final analysis included the wheel gyroscopic effect on a shaft which further raises the critical whirling frequency. This was only included in the final analysis (d), to determine more closely the value of the critical frequency in operation. This is shown by curve (e) in Fig. 21. The range of likely bearing flexibilities shown in Fig. 21 was chosen by reference to the calculated whirling speeds of the original rotor and the speed range over which vibrations had been measured.

Vibration measurements carried out by the shipyard indicated a reduction from about 20 mm/sec (R.M.S.), at 6500 r.p.m. for the original rotor to a maximum of about 2·0 mm/sec (R.M.S.), after the design modifications had been incorporated.

**Case 2. Oil Tanker. 250 000 dwt. Delivered 1972. LR Class.**

The cargo pump manufacturer reported that excessive vibration had caused the C.O.P. turbine overspeed trip to operate at the normal pump service speed of 1300 r.p.m., and that because it had been suggested that torsional vibrations could, in some way, be a contributory factor, the manufacturer requested that the Society undertake a torsional vibration analysis.

Plans of the shafting system were submitted by the manufacturers, together with the relevant mass-elastic data. The equivalent mass-elastic model representing the shafting system is indicated in Fig. 23(a).

The first natural torsional frequency found was 741 c.p.m., and the second was 5733 c.p.m. Entrained fluid was taken to be 25 per cent of the impeller inertia, but since neither this figure nor the mass-elastic data supplied were considered to be exact, the calculated resonant frequencies may have been in error by some $\pm 20$ per cent.

Figs. 23(b) and (c) illustrates diagrammatically the torsional swinging forms of the system for each of the calculated natural frequencies.

The first natural torsional frequency could not be excited at 1300 r.p.m. even allowing for a $\pm 20$ per cent error in calculation. The second frequency could be excited by a fourth order vibration at the maximum speed, but as seen from the swinging form in Fig. 23(c), the maximum amplitude occurs at the upper Hooke's joint of the flexible drive shaft. It was concluded that torsional vibrations would not be likely to cause the turbine overspeed trip to operate, and recommended that the rotor critical whirling speed characteristics be examined.

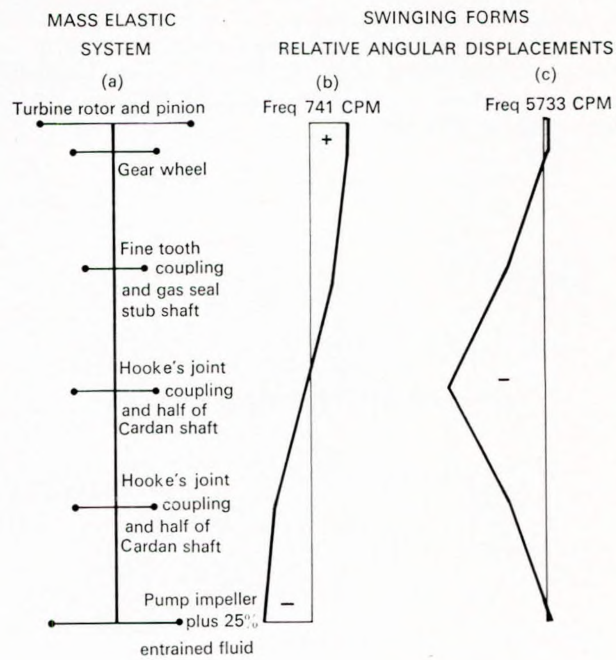



FIG. 23

(a) Representation of the mass elastic system. (b) & (c), Calculated torsional swinging form modes, and angular displacements.

**Turbine Blade Vibration Analysis**

More recently one cargo pump turbine manufacturer had requested the Society to undertake a computer-aided finite element analysis to determine the blade packet resonant frequencies and the dynamic response characteristics of two prototype blade designs, and one existing blade design.

The finite element technique embodies the 'lumped element' approach, wherein the distributed physical properties of the blade packet structure are represented by a mathematical model consisting of a finite number of idealized substructures (or elements) that are interconnected at a finite number of grid points, to which loads may be applied directly or through offsets, and to which various kinds of constraints may be applied.

The element is a convenient means for specifying many of the properties of the structure including material properties, mass, damping, and stiffness. Each grid point can have six degrees of freedom, viz. three rotational, and three linear motions.

In this instance each of the 29-blade packets was modelled on cylindrical co-ordinates to simulate the curvature of the wheel, and each blade and shroud pitch was idealised by 21 bar elements.

Perhaps the most significant feature of this particular analysis was the development of the facility to include the coupled bending-torsion modes. The blades were tapered from base to tip, but each section was axi-symmetric, and by adaptation of the multi-point constraint method for the axi-symmetric section the use of the relatively simple bar element was possible, the grid points being located along the locus of the twist centre, and the mass centres being offset along the axis of symmetry.

A total of some 75 different resonant frequencies was found by analysis within the range of possible first order exciting frequencies. Plots of three of the normal mode swinging forms for the 29 blade packet are illustrated in the accompanying diagram (Fig. 24) showing plan and side views of the same packet of blades when vibrating in (a) the tangential in-phase mode at 3299 Hz, (b) the tangential out-of-phase mode at 10 020 Hz, and (c) the axial out-of-phase mode at 5951 Hz.

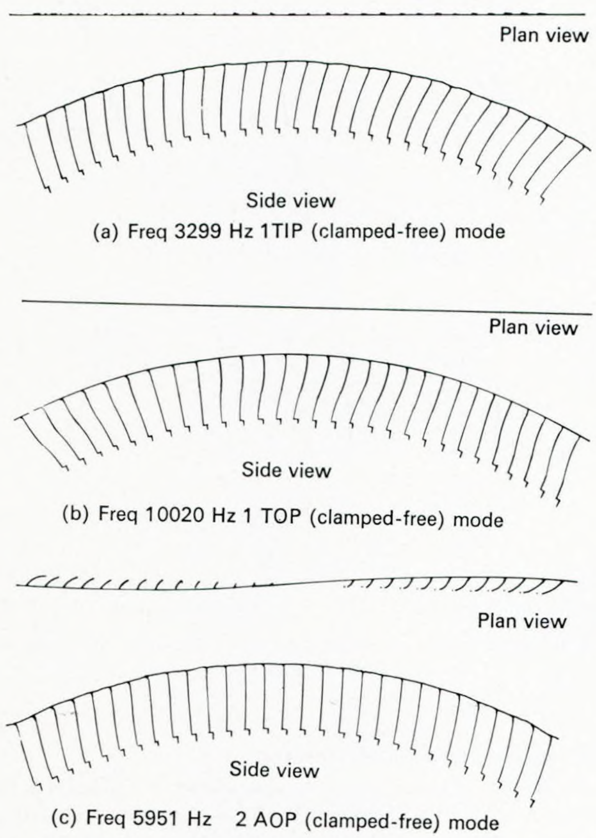



FIG. 24

Plan and side views of the plotted swinging form modes of a packet of 29 turbine blades.

Finally a forced damped dynamic response analysis was undertaken on 64 resonant frequencies to determine the vibratory stress in blades, tenons and shroud. The results indicate high levels of vibratory stress when the turbine is operating at maximum speed and will probably necessitate a design modification.

## CONCLUSIONS

Very little has been written on the subject of cargo oil pump machinery installations, or of the various problems associated with their design or operation, yet my direct encounters with Shipyards, Owners, and Manufacturers have led me to conclude that this is a subject which could benefit from discussion.

The time limit does not permit a more detailed treatment of all the component parts of the machinery, but the range of the study is a fair reflection of the problems areas in which the R & TAS Department has been directly or indirectly involved in recent years.

## ACKNOWLEDGEMENTS

The Author would like to thank his colleagues in the Research and Technical Advisory Services Department for their valuable help, and to the Committee of Lloyd's Register of Shipping for their permission to publish this paper.

He would also like to thank the following for providing the illustrations namely: —

Jönköpings Mekaniska Werkstads A.B.
Jönköping, Sweden.

Peter Brotherhood Ltd. Peterborough, England.

Shinkokinzoku Industries & Co. Ltd. Japan.

Stal-Laval Turbin A.B. Finspong, Sweden.

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DISCUSSION

MR H B CRAWFORD (Weir Pumps Ltd) said that the author had sought to analyse many of the problems associated with the primary equipment for handling the discharge of the cargo in oil tankers.

Many of the points he raised had been the subject of considerable development by his own company, through the period described in the introduction of the paper. The company, in its present and earlier form, as E & J Weir Ltd and Dysdale, had been associated with Marine Engineering for over 100 years (1874).

For many years, they had manufactured centrifugal pumps for cargo oil handling and the turbines for driving them. The author referred to the recently introduced barrel design, but pumps of this type had been in service for some 30 years and a brief note on the effect of commercial pressures on pump design with special reference to the barrel casing type would not be out of place.

In the late 1940's when the centrifugal pump began to replace the duplex pump for cargo oil discharge, Drysdale and Company had introduced the vertical barrel casing cargo oil pump to the market. The pump was designed initially to meet the requirements of Shell and the first units were installed in the VEXILLA and VIBEX. Hundreds of pumps of this type were subsequently installed.

This was the true vertical cargo oil pump design with overhead bearing assembly and the impeller overhung in a casing split either radially or axially. There was no external bottom bearing. This design allowed the impeller and suction branch to be in the lowest position for best possible NPSH and suction performance.

In the 1960's, to meet the growing international competition, particularly from Japan, the market became increasingly price conscious and less specification conscious. This led to tne introduction and market acceptance of the "adaptable cargo oil pump". In vertical installations this meant taking a horizontal axially split pump with outboard bearings and placing it in the vertical attitude. This put the suction branch in a high position relative to the base of the pump but also, as the author had indicated in his paper, placed the external bottom bearing in a situation not readily accessible for routine lubrication and maintenance, and also in an environment conducive to neglect leading to possible trouble and bearing failures.

In taking up this point in the paper, the submission was that the external bottom bearing in the vertical axially split cargo pump was not there by design but rather by commercial pressures in a competitive market.

It should be noted that up until the early 1970's, his company was, with the possible exception of K.S.B. the only company to have the vertical barrel casing overhung impeller design.

The faith in the true potential of the barrel design was borne out when in the 1970's, as the author had indicated, there was a re-emergence of the requirement for barrel casing pump for installations not only in large crude carriers but also in product tanker tonnage.

In 1970 a radical change in turbine design philosophy took place within the company so that their products would meet the changing market demands. The trend towards increased automation leading to computer control both in the engine room and the cargo handling equipment. The

W.P.L. response was the introduction of the A.E.T. Turbine (the Adaptable Electronic Turbine). Adaptable: mountable either vertically or horizontally; Electronic: electronic control and monitoring.

This range of units was designed to be simple and robust. To achieve this, all gearing other than the primary train was eliminated. The control of the turbine was through a speed sensor viewing flats on a collar on the shaft (Fig 25, turbine) between the turbine rotor and the integral pinion gear on the shaft, thus meeting the desiderata set out by Mr Donald.

It should be noted that while this probe was so arranged that it fed impulses into the turbine controls and overspeed circuit and was inherently fail safe, a mechanical overspeed ring governor was provided as a back-up while the integrity of the electronic trip was proved to the satisfaction of Lloyd's Register of Shipping and other Societies. The control system fed a signal to the combined steam throttle valve E.S.V which, because it was in constant motion was free to close rapidly when required to do so.

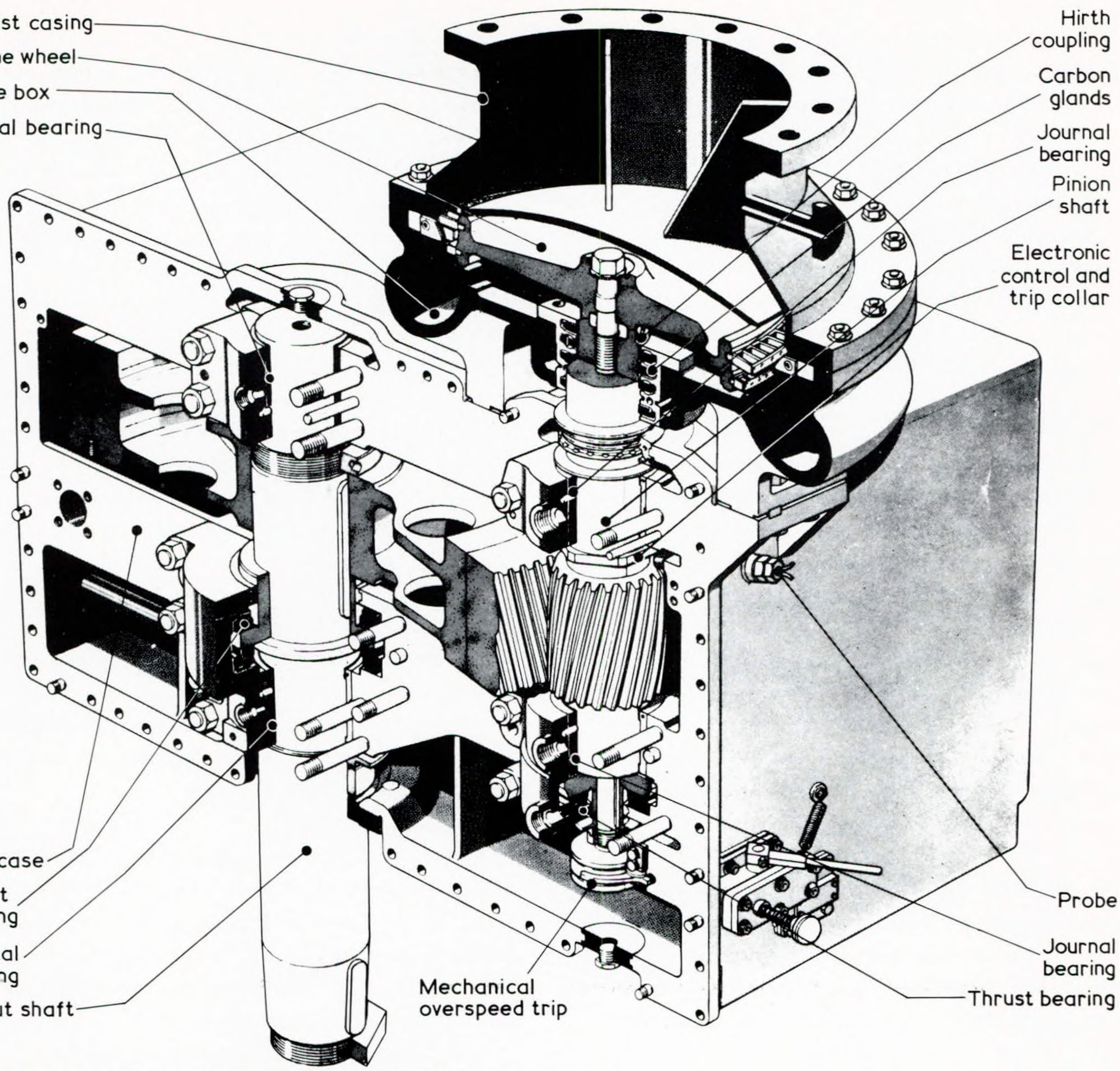
The turbine was further designed with either a short or a long shaft extension on the output shaft. The short shaft was intended for horizontal drives with a conventional bulkhead seal, while the long shaft was used for vertical installations when the shaft extends through a special deckhead seal (Fig 26, seal) into the pump room resulting in a more compact arrangement, i.e. returning to the three element shaft system as used in horizontal installations.

In the design of those turbines, safety was an important criteria and the use of cast iron in the flake form was eschewed because of its lack of ductibility and tensile strength to contain either the loss of blades and/or a rotor burst should the controls be gagged. Shrunk-on pinions had also been avoided because of the stress raiser at the end of the shrink, and the effects of the collar on a deflecting shaft alluded to by Mr Donald. However, shrunk-on thrust collars for the axial location bearing beyond the second journal bearing on the shaft had not proved any problem either as to shaft integrity or critical speeds.

The turbine rotor, a simple single web design with a robust rim supporting two rows of blades had proved satisfactory provided that careful analysis of the stress pattern in the web was made. The attachment of the rotor to the pinion shaft must be positive and have low lateral flexibility, thus rather than the flanged connection typified by Fig 14, a hirth, i.e. toothed coupling, was used together with a central high tensile bolt. This arrangement gave accurate location of the rotor through all thermal transients, and the attachment point was close to the mass centre of the rotor so that bending effects on the joint were minimised.

Turbine blade failure was a problem which had exercised most designers, and the author had correctly raised the problem of blade resonance both as a single unit and in batches. It had been Mr Crawford's company's experience that the use of the fir-tree method of retaining the blades in the rotor had been a reliable one, provided that the section between the blade proper and the fir-tree root was correctly designed, using photo-elastic models as well as calculation. The above would produce a blade fixture of high strength and fatigue resistance. The situation could be further improved by well riveted shroud banding spanning the correct number of blades, thus raising the blade batch resonance frequencies in the modes illustrated in Fig 24 to levels where blade failure was not encountered, there being no corresponding synchronous generators.

The author had also raised the problem of




Standard rotation: anti-clockwise, looking on output shaft coupling

Fig.25 – Turbine

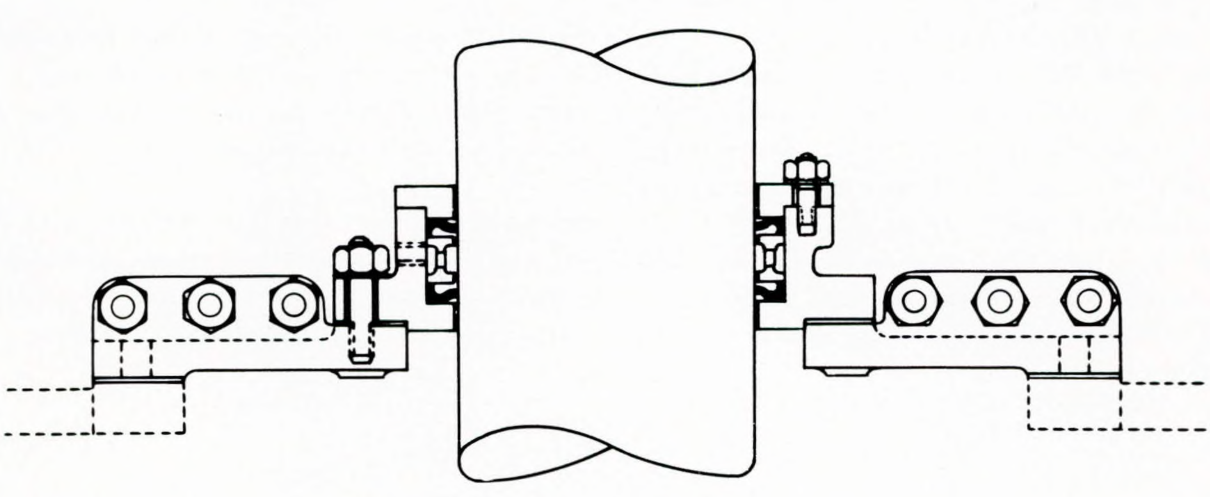


**Fig. 26 – Vertical deckhead gas seal**

torsional response of the drive train. This was most pertinent in vertical installations and, since W.P.L. often supplied both the pump and driver, it was easier for them to specify the requirements for the intershaft. Once the intershaft had been selected and its characteristics known, analysis of the characteristics of the drive train in both the transverse and torsional modes could be carried out. In this, the effect of a wet or a dry pump impeller could be noted.

Experience had shown that it was good practice to quote the pump speed corresponding to the turbine overspeed setting as being the design speed for the intershaft. This resulted in the selection of a more robust intershaft and once the train was assembled, no critical speed problems either torsional or transverse had been encountered. It was believed that this was because of the ability to analyse the system as a whole, and the "correct" specification of the duty for the intershaft.

Being able to test the whole system it had been possible to check the theoretical predictions against actual behaviour. At the same time they had been able to assess the stresses in the vibrating assemblies particularly for the vertical installations using barrel casing pumps. Results had shown close correlation between theoretical production and actual: the discrepancies being of the order of 1 Hertz in 30. Tests had also borne out the twice rotational speed excitation caused by the Hooke's joint on the intershaft, however, the cyclical stresses set up had been low in his company's installations. Using the ASME VIII Division 2 fatigue life criteria, both intershaft and pump shaft, which proved to be the most highly displaced components, in the first natural mode, had life expectations well in excess of $10^6$ cycles.

In practice it could not be too highly stressed that any drive system involving splined shafts, gear type couplings or other devices where axial movement would be encountered, must be regularly lubricated or fitted with a long life, self sustaining, lubricating system to prevent seizure or gumming which, if not corrected, could lead to high axial loads during the flexing of the vessel, either at sea or during discharge of the cargo.

Finally, the paper had raised many matters of vital concern to those supplying, installing and operating cargo pumping equipment and the distilled experience of an organisation such as Lloyd's Register on these matters was invaluable.

MR S SPEED FIMarE (BP Co Ltd) found the paper very interesting and agreed with Mr Donald's concluding remarks regarding faults in design, application and operation of these very large pumps.

The speed controls, governing, etc., of these units was a most important feature and while there had been great improvements in design there were still weak links in the system. A fault had occurred in two units of one of his company's vessels resulting in the total disintegration of the turbine. In both cases, the seat of the steam control valve, which was welded into the body of the valve, became detached, and overspeed resulted.

Could the author comment on any problems of line-up of vertical and horizontal cargo pumps due to flexing of the ship in cargo handling operations ?

The balancing of flexible drive shafts on vertical cargo pumps, sometimes presented a problem. Did the author have any experience or views on balancing of these shafts in situ ?

The author had mentioned fine tooth flexible couplings used in a lot of cargo pump drives. What were his views on the suitability of this type of coupling in this application, and was it in fact flexible at all under conditions of full load ?

CORRESPONDENCE

MR F S LYNAM (Esso International Services Inc) wrote saying that for many years, cargo oil pump turbines were of such low power that they were almost inevitably over-designed for their normal duty. To make them otherwise would have made them too expensive and too much of a "watchmaking" job for marine use. Also, their steam consumption was not important because the total pumping power installed per vessel was low, relative to the installed boiler capacity. The other parts of the pump drive system were similarly robustly proportioned.

But tanker size increases culminating in the introduction of VLCC in the late 60's produced changes as follows:

1) Powers increased. Therefore, the turbine manufacturers had an economic incentive to make their machine run nearer to its design limits.

2) Steam demand became of increasing importance, especially with large slow-speed diesel-engined tankships where the cargo pumping installation's demand determined the installed boiler capacity.

3) Vertical installations were favoured primarily because of their compactness, so long drive shafts were required.

As indicated by the author's experience, some of these higher power installations were not analysed in sufficient depth at the design stage. But some manufacturers did provide adequate installations which have given good service.

Torsional vibrations had not usually been a problem but Mr Donald's remarks on whirling speeds were very apt. The linked-shaft often fitted between the deck-seal and the pump had negligible damping, and damage soon resulted if its whirling speed was reached. It was not sufficient to keep the true first-lateral critical speed above the tripping speed, because tripping in service would occur during a rapid and unscheduled acceleration of the unit and the maximum speed attained would be higher. With tripping at 15% overspeed, the whirling speed should be at least 20% above normal running speed.

Balance of the linked-shaft was also most important. Furthermore, it could not be treated as a rigid element, relative to centrifugal forces, unless it was balanced at every position along its length. An acceptable degree of multi-plane balancing was therefore desirable. If a torque tube had circumferential wall-thickness variations changing along its length, then controlled bending in the plastic range might be one way of improving its dynamic balance.

The author had commented at some length on the turbines' design features. These machines had an arduous duty to perform which was not always fully considered by their designers. The following points were noteworthy:

i) Power absorbed by the pump would be greatest when pumping sea water at full rated speed. This could cause cavitation damage to the pump impeller in some circumstances and ballast water movement was often effected at reduced pump speed to alleviate this problem. For this and other operational reasons, the pumps operated regularly in the 80 - 100% speed range.

ii) Whether pumping cargo oil or sea water, the pump would lose suction on some occasions. Then the speed would increase and the turbine governor would normally regain control. But the turbine was then developing little power and its rotor blades would receive steam at a temperature not far below that of the inlet steam. Apart from the thermal shock involved, the blades and shrouding would be temporarily weakened by temperature at a time when the turbine might be running into the overspeed region.

iii) After losing suction, the fluid might return suddenly after a short time giving a mechanical shock to the drive-line.

iv) The machines suffered long periods of inactivity between successive periods of operation. Their wheel-casings were then subject to the engine room's atmospheric conditions. Corrosion could be a result.

v) Long runs of steam pipe with sections having low slope often preceded the turbine's stop valve. Even with care, line drainage could be a problem, especially during the heating-up phase.

It was apparent that turbines should be designed with increased safety factors for these conditions, especially where superheated steam was employed.

It was beneficial to have an effective vacuum-priming and pump control system available for use during conditions when loss of suction might occur. However, even these devices could provide little protection to the drive-system if wax or mud from the cargo suddenly ingested. Some grades of crude oil were particularly prone to separation of deposits. One drive system failure investigated was associated with mud being suddenly carried into the cargo pump, despite the presence of a pump section strainer in good condition. The pump casing was found to be about half full of a deposit with the appearance and consistency of black shoe polish (see Fig 27).

Such extreme conditions were infrequent but similar less severe disturbances were part of the normal life of a cargo-pumping installation.

Instrumentation of good quality was also important. An error of 20% in a pressure gauge reading or 10% in a tachometer reading could lead to low or high flow rates which might be first manifest by pump casing overheating or less of suction respectively.

AUTHOR's REPLY

The author thanked Mr Crawford for his interesting contribution representing the combined experience of two well-known pump manufacturers.

Mr Crawford had mentioned the new design of COP installation his company had introduced in 1970, which appeared to have avoided the pitfalls outlined in the paper. It was probably significant that safety was considered an important criteria in the design conception.

The author agreed with Mr Crawford's remarks about turbine blade retention by the fir-tree root method. It would probably be one of the more expensive methods of blade attachment, but certainly one of the most efficient if properly designed, manufactured, and fitted.

It was always encouraging to find an independent but mutually reinforcing correlation between the observed and the theoretically predicted behaviour of vibratory systems. He supposed that Mr Crawford meant a discrepancy of $\pm$ 1 Hz in 30 between predicted and observed frequencies, which was probably a reasonable correlation.

In reply to Mr Speed the author did not have any figures to quote for the effects of hull deflections upon the alignment of cargo oil pumps. The recommended alignment and other measurements which the Society had offered to carry out, as mentioned in Section X3, in the last paragraph, had not actually been taken up by the client.

Everyone agreed that changes in draught conditions produced hull deflections but he had not met anyone who could quote figures for the relative movement between pumproom flat and pumproom deck head. He would have thought that the only significant double bottom deflection would be that caused by increased hydrostatic pressure in the loaded condition which could upset the parallel shaft alignment requirement for the vertical installations, particularly for the outboard pumps, which could be set over at a greater angle than the inboard pumps, but in view of the close proximity of the two pumproom bulkheads, he thought it unlikely to be a significant factor in pump and gearshaft alignment. More to the point was the correct alignment of the initial installation and avoidance of any local temperature gradients and associated effects which could change the initial alignment.

Horizontal arrangements of pump and driver posed a somewhat different problem because of the gas-tight bulkhead seal in way of the shorter cardan shaft. Any vertical or transverse movement of pump or gear-shaft axis would impose a transverse load on the cardan shaft in way of the gas seal.

In the author's opinion, it should not be necessary to have to start balancing the flexible drive shafts in situ, for it was to be hoped that this had been undertaken by the manufacturer under properly controlled conditions prior to despatch from the works. Again, should an installed shaft develop an unbalance condition in service, a prudent owner would first establish the cause of the unbalance before proceeding with rebalance. In that case, one method which had been used in balance of torque tubes on horizontally arranged machinery is to attach seismic pick-ups in the vertical and transverse planes on the bearing keeps of the adjacent bearings at each end of the cardan shaft, (or torque tube). By fitting a sine-wave generator at a free end of the shafting system it would then be possible to determine the phase relationships. By attaching known weights at prescribed angular positions on the cardan shaft a quantitative relationship would be established between unbalance weight and vibration velocity reading for various angular positions, and thus the unbalance quantity determined. Mr Lynam had succinctly outlined the problems associated with cardan shaft unbalance, but the author hesitated to endorse Mr Lynam's recommendation for controlled bending in the plastic range to achieve an improvement in its dynamic balance.

Regarding the use of fine tooth couplings for cargo pump drives, he could only say that his attention had not been drawn to any failures up to the time of writing the paper, although such couplings in other types of installation did occasionally come to his notice. In most instances, the problems related to (a) deficiencies in material, manufacture, or subsequent heat treatment, (b) loss of, or inadequate lubrication, (c) operation under conditions of alignment in excess of the allowable, or (d) a combination of the causes outlined.

Whether it was suitable for cargo pumps drives would depend upon the magnitude of changes in alignment due tc hull deflections as previously referred to but he thought that when transmitting full load torque the fine tooth coupling would tend to lock solid if the angular displacement between driving shaft and cardan shaft were nil or very small. In that case, if an axial expansion in the shafting occurred, the coupling would be likely to take up the expansion in a stick-slip fashion and be released in jumps, whereas if the correct amount of angularity existed between shafts he felt the expansion would be taken up more smoothly. The problem was to know where to set the shafts during installation so that optimum angularity was achieved at full torque.

There was a tendency to think of flexible couplings as self-righting from all starting points. This was not so. They required careful selection for the type of duty expected and careful alignment, and maintenance.

He would like to thank Mr Lynam for his most informative and important contribution to the discussion from the operator's point of view.

He agreed unreservedly with Mr Lynam that the linked-shaft fitted between the deck-seal and pump should be designed to ensure that the lowest whirling speed should be well above the maximum speed which it could attain. The figure of 20% above the operating speed for a 15% turbine overspeed trip speed sounded reasonable but of course it would depend upon the accuracy with which the lower whirling speed of the installed link-shaft could be predicted, making due allowance for bearing stiffnesses.

Mr Lynam had stressed the importance of good quality instrumentation, and there was no doubt in the author's mind that this was sound advice, but probably equally important was the need to maintain the instrumentation in good condition and to carry out frequent calibration checks. For example, in the context of the whirling speed of the linked shaft mentioned above, if the 15% turbine overspeed were set using a tachometer which was reading 10% low, a linked shaft whirling speed which might otherwise have been outside, would now be within the overspeed range.

# THE INSTITUTE OF MARINE ENGINEERS

## TRANSACTIONS

# PRESIDENTIAL ADDRESS

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*To be presented at the Memorial Building, 76 Mark Lane, London EC3, on Tuesday, 3 October 1978, at 17.30*