A Study to Assess the Energy Savings Potential

in the Ocean Going Trawler "Roxana Bank"

by

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<u>DURBAN</u>

I hereby declare that the dissertation represents my own work, both in conception and execution.

<u>D M FIDDLER</u>

<u>DURBAN</u>

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SUMMARY

Increasing fuel prices have forced marine engineers and diesel engine manufacturers to look at methods of reducing fuel consumption without a loss in output power.

Engineers are always investigating the following points:

(1) decreasing the specific fuel consumption

(2) enabling engines to use worse fuels

(3) extending part load capability

(4) using as much waste heat as possible.

The sources of heat loss are investigated and the design of an efficient heat recovery system has been evaluated for ideal temperature and heat conditions, after taking into consideration the various methods of heat recovery that are possible on the fishing trawler MFV Roxana Bank.

Sources of heat loss identified by engine manufacturers are found primarily in engine cooling water and exhaust systems. These methods are investigated and extra heat transfer systems have become apparent.

The use of exhaust gas boilers in diesel engine installations has become widespread as there is always a demand for heating services regardless of vessel type.

The predominant form of heat transfer on the Roxana Bank is by forced convection and fluids must always remain in a turbulent state when passing through heat exchange apparatus.

The introduction to heat exchangers shows that a high degree of turbulence in both primary and secondary fluids will the overall heat transfer coefficient and also reduce fouling.

The design parameters of the heat recovery systems discussed as suitable for the Roxana Bank require the existing cooling water system to be scrapped and a fresh water system with one central cooler to be designed with a high temperature and a low temperature circuit together with a variable speed pump arrangement.

The central cooler system is discussed in detail and it is shown that existing ships and/or new buildings should be fitted with such central cooling systems for a number of reasons.

The heat recovery system requires a certain degree of automation to prevent undercooling of the main engine systems and to maintain set inlet and outlet temperatures to makers specifications.

After attending the Roxana Bank on the 09/10 November 1989, the monitored operating parameters show a marked deviation to the ideal temperatures, particularly in the cylinder cooling water circuit temperatures. The adverse effects these deviations have on the heat recovery and combined engine load fluctuations found on the Roxana Bank show that steady state heat recovery is difficult to achieve on this particular type of vessel.

Through calculation it is shown that the envisaged steam turbine alternator is not practical due to the combination of low feed water inlet temperatures to the boiler and insufficient driving steam. The consequence of the reduction in electrical load on the power take off alternators have on the exhaust gas mass flow rates due to reduced engine load are also discussed.

A brief mention is also made of organic fluid and thermal oil systems as a form heat recovery and possibilities for efficient heat transfer using these systems.

Recommendations are made to Irvin and Johnson with regards to a centralised cooling system and the benefits that can be realised with regard to more efficient heat exchanger operations, but that heat recovery on the Roxana Bank from the proposed sources is not a viable proposition.

OPSOMMING

Stygende brandstofpryse dwing marine-ingenieurs en vervaardigers van dieselenjins om metodes te ondersoek waardeur brandstofverbruik verminder kan word sonder verlies aan dryfkrag.

Ingenieurs ondersoek gewoonlik die volgende hoofaspekte:

(1) Vermindering van spesifieke brandstofverbruik.

(2) Aanpassing van enjins om op swakker brandstof te loop.

(3) Uitbreiding van gedeeltelike vragvermoë.

(4) Aanwending van soveel afvalhitte as moontlik.

Die bronne van hitteverlies word ondersoek, en met inagneming van die verskillende metodes van hitteherwinning wat op 'n seevarende treiler moontlik is, word die ontwerp van 'n doeltreffende hitteherwinningstelsel ten opsigte van ideale temperatuur- en hittetoestande geëvalueer.

Volgens enjinvervaardigers vind hitteverlies hoofsaaklik in die waterverkoelstelsel van die enjins en in die uitlaatstelsel plaas. Metodes om die verlore hitte te herwin word ondersoek en 'n behoefte aan bykomende verhittingstelsels kom aan die lig.

Uitlaatgasketels in dieselenjin-installerings word reeds algemeen gebruik en daar is altyd 'n aanvraag na verwarmingsdienste, afgesien van die soort vaartuig wat betrokke is.

Die belangrikste vorm van hitte-oordrag op die ROXANA BANK is met behulp van gedwonge stroming en vloeistowwe moet dus altyd in 'n turbulente toestand gehou word terwyl dit deur hitte-uitwisselaars vloei. Die inleiding tot hitte-uitwisselaars toon dat 'n groot mate van turbulensie in albei vloeistowwe die algemene hitte-oordragkoëffisiënt verbeter en ook die opbou van ketelsteen verminder.

Die ontwerpsparameters van die bespreekte hitteherwinningstelsels wat vir die ROXANA BANK geskik sou wees, vereis dat die bestaande waterverkoelstelsel vervang word deur een sentrale varswaterverkoeler met 'n kringloop van hoë temperatuur en lae temperatuur, asook 'n pompstelsel met

wisselbare spoed.

Die sentrale verkoelstelsel word in besonderhede bespreek en daar word getoon dat bestaande skepe en/of skepe in aanbou om verskillende redes met hierdie verkoelstelsel toegerus behoort te word.

Die hitteherwinningstelsel vereis 'n sekere mate van outomatisasie om onderverkoeling van die enjin te verhoed en die inlaat- en uitlaat-temperature te handhaaf wat deur die vervaardigers gestel is.

Op 9 en 10 November 1989 is 'n ondersoekvaart op die ROXANA BANK onderneem en daar is gevind dat die gemoniteerde bedryfsparameters aansienlike afwykings van ideale temperature getoon het, veral wat betref die waterverkoelingskringloop vir die silinders. Die nadelige uitwerking van hierdie afwykings op hitteherwinning, gekombineer met die wisseling in enjinlading wat op die ROXANA BANK opgemerk is, toon dat bestendige hitteherwinning feitlik onmoontlik is.

Berekenings toon dat die beoogde stoomturbine-alternator nie prakties is nie, vanweë die kombinasie van lae watertemperatuur by die ketelinlaat en ontoereikende aandryfstoom. Die uitwerking van die verlaging van die elektriese lading van die kragaftakker-alternators op die uitlaatgasmassavloeitempo word ook bespreek.

Organiese vloeistof- en termiese oliestelsels as 'n vorm van hitteherwinning word kortliks genoem.

Aanbevelings word aan Irvin en Johnson gemaak aangaande 'n gesentraliseerde verkoelstelsel en die voordele wat dit inhou, maar hulle word meegedeel dat hitteherwinning nie prakties uitvoerbaar is nie.

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THE PROBLEM AND ITS SETTING

1. The Statement of the Problem

The purpose of this investigation is to evaluate the sources of heat (energy) loss and evaluate their application potential in terms of operating efficiency in the ocean going stern trawler ROXANA BANK.

1.2.1 <u>The First Subproblem</u>

The first subproblem is to identify and evaluate the sources of heat from the main propulsion engine and refrigerating plant and relate the data to specifications for the design and evaluation of the feed water system to increase the output of the exhaust gas boiler.

1.2.2 <u>The Second Subproblem</u>

The second subproblem is to select a suitable turbine that is compatible with the steam output of the boiler and to evaluate the output power available for electrical generation in terms of cost effectiveness and operation.

1.2.3 <u>The Third Subproblem</u>

The third subproblem is to relate the results concluded from the data in subproblems one and two to the cost effectiveness of the proposed system and predict the performance against the existing operating criteria in the ROXANA BANK.

1.3 The Hypotheses

1.3.1 Hypothesis One

The first hypothesis is that the measurement of energy losses can be converted into specifications for the design of an efficient feed system to increase boiler output.

1.3.2 Hypothesis Two

The second hypothesis is that the compatible steam turbine will be cost effective and it will be possible to demonstrate the cost effectiveness.

1.3.3 Hypothesis Three

The third hypothesis is that the results obtained will show that the cost effectiveness can be related to operating efficiency of the ROXANA BANK.

1.4 The Assumptions

1.4.1 The First Assumption

The first assumption is that the owners of the ROXANA BANK will not affect any changes to the main propulsion engine during the course of this study. .

1.4.2 The Second Assumption

The second assumption is that the engine test bed data will be valid for the comparative studies and calculations.

1.4.3 The Third Assumption

The third assumption is that the ROXANA BANK will always be employed in the same trade, in other words, bottom trawling operating out of Cape Town in the traditional South African fishing zone.

1.5 The Delimitations

1.5.1 The study will not evaluate any fuel saving systems on the main propulsion engine.

1.5.2 Actual data will be used from the engine room log books and from temperatures and pressures compiled during vessel operation monitoring periods.

1.5.3 The study will not include the design of the steam turbine and the alternator.

1.5.4 The study will accept that the exhaust gas oilfired combination boiler is efficient.

1.5.5 The study will be limited to the sister ships ROXANA BANK and ROSALIND BANK.

1.5.6 The engine builders (SULZER Brothers) test bed operating specifications, graphs and charts will be deemed correct and will be used in calculations.

1.5.7 Evaluation of cost effectiveness of the proposed plant to the existing plant will be based on fuel consumption only. An estimate for the installation of the proposed plant will be given and will not reflect manhours taken.

1.6 The Importance of the Study

In South Africa the large fishing companies have two types of trawler that can be divided into the following categories:

Category A: The "wet" fish type that remains at sea for 5 to 7 days and stores the catch in ice. The fish is not frozen. Examples of this type of fishing vessel are MFV's Arum, Begonia, Aloe, Gardenia.

Category B: The "freezer" trawlers that stay at sea for 40 to 50 days and do all the processing of the fish on board. The fish is then frozen in plate freezers and stored in refrigerated holds maintained at approximately

-25°C. Examples of this type of vessel are the MFV's

Roxana Bank, Rosalind Bank, Storesse.

Many of these trawlers of both categories have been attended in Cape Town by ship surveyors and marine engineers and have undergone surveys of various kinds ranging from safety equipment surveys to machinery inspections. During discussions with engineer superintendents and management of these trawler companies (Irvin and Johnson, Sea Harvest) the fact has emerged that there is concern over fuel consumption and fuel prices and methods of saving fuel. High fuel prices contribute to high operating costs.

Various factors contribute to increases in operating costs of which fuel prices and inefficient running of engines contribute. An example of the fluctuation in fuel prices can be found in the FAIRPLAY magazine where the fuel price for heavy fuel oil (180cSt) was 60 US dollars per metric tonne (date 14 Jan 1989) and the given fuel price in the same publication dated 14 November 1989 quotes a figure of 104 US dollars per metric tonne.

The trawler companies have been using marine diesel oil but as prices rose two options were considered:

(1) Running the main propulsion engine on heavy fuel oil and the auxiliaries on diesel fuel oil.

(2) Running the main propulsion engine on blended fuel. (Blend is approximately 82% heavy fuel oil and 18% marine diesel oil).

Running the main propulsion engines on heavy fuel oil has never been particularly successful for the size of engines and the most success has been found using blended fuel although there are limitations to this type of fuel as well.

The second factor which affects the efficiency of operation is the energy loss through the continuous running of the main propulsion engine. Based on the assumption that the main engine is running at approximately 85% of maximum continuous rating, the cylinder exhaust temperatures are in the region of 450°C with the outlet temperature from the exhaust gas turbocharger in the region of 380°C.

Further examples of energy loss on the main engine is through the jacket fresh water cooler, lubricating oil cooler and the charge air cooler. Another energy loss separate from the main propulsion unit is through the cargo refrigeration freon gas condensers.

In all cases the majority of waste heat is lost up the exhaust pipe to atmosphere or directly to the sea via the heat exchangers. The trawler owners are aware of these losses and have carried out modifications to improve operating costs. These modifications are more of a mechanical nature as they are restricted in their investigations and efficiency research by lack of personnel to carry out such investigations, and they are essentially ship operators and not ship builders.

There have been attempts to date to harness this waste heat and this study will investigate through design the harnessing of energy losses relating to increased plant efficiency and cost effectiveness.

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CHAPTER TWO

THE REVIEW OF THE RELATED LITERATURE

The prospect of improving overall fuel efficiency in the propulsion engine itself is beginning to approach the feasible limits. Therefore, more effort is being put into exploring methods of profiting from available surplus energy. (Anon 1988b: 44)

Morton (1981:16) has identified that the heat available from a diesel engine comes from three main sources:

(1) Exhaust gas at an initial temperature 300 - 350°C.

(2) Charge air at an initial temperature of 150 - 200°C.

(3) Water or oil for pistons, jackets for cooling up to 75°C.

The survey carried out (Anon 1988: 44) confirms the temperatures discussed by Morton.

Gallois (1981:9) has carried out calculations and determined that the engine cooling water outlet temperature can be as high as 130°C with the engine inlet temperature at 120°C. Operating at these temperatures allows a feed water temperature of 120°C into the boiler. The system, employing a SEMT Pielstick PC4 medium speed engine also incorporates a heat recovery unit in the charge air cooler raising the feed water temperature.

The energy losses to exhaust and cooling water are illustrated on a Sankey diagram and reading figure

2.1 Sankey diagram showing energy losses - the loss to cooling water and exhaust gas is quite substantial.

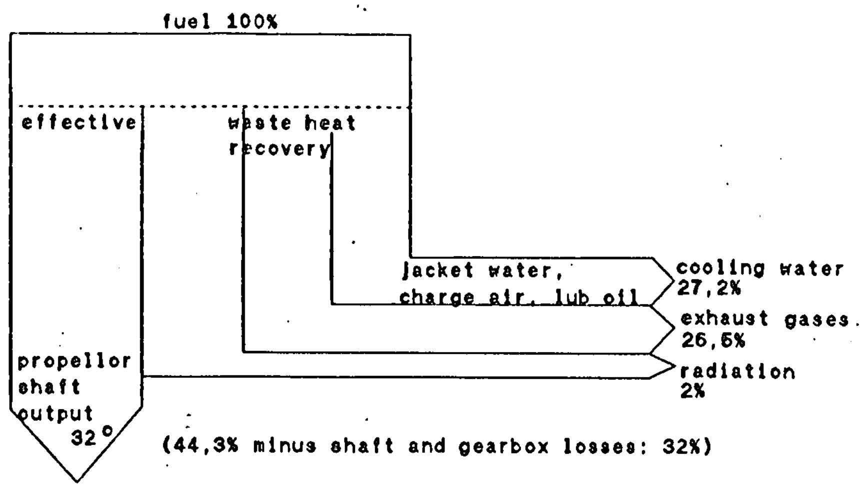


Figure 2.1 Sankey Diagram Showing Energy Losses.

As already stated, to improve plant efficiency the sources of heat loss must be evaluated as to the effectiveness of heat recovery and the overall suitability of the system.

If the primary objective is power recovery, high grade exhaust heat should be kept for this purpose rather than heating duties. (Anon 1988b:44). Gallois (1981:9 & 10) has calculated that 9 to 10% of the main engine power can be recovered using the system recovering heat from the charge air cooler and operating the jacket water at 130°C. He has concluded however that the heat recovery capacity which enables an exhaust gas boiler to supply the entire electric demand is possible only if the propulsion power is above 6000 kilowatts. Gallois (1981:9) also concluded that the electrical demand for a non refrigerated ship of 7500 kilowatts is in the region of 4% of maximum continuous rating of which 30% is required for engine auxiliaries, fans and lubricating oil pump. Fan capacity can be reduced by up to 80% by leading the turbocharger suction to outside the engine room.

When reducing electrical load in a project for heat recovery and energy saving, the engine type plays a very important part in the recovery system. Reducing electrical load on an engine fitted with a power take off coupled to an electrical generator will actually reduce the load on the engine and hence reduce exhaust gas mass flow rates.

A system fitted with an engine supplying propulsive power only will benefit more as electrical load reductions will reduce the load on the auxiliary engine and have no effect on the main engine operations.

The exhaust temperature after the heat recovery unit should not be lower than 170° to 180°C (Gallois 1981:9) to prevent sulphuric acid attack.

As the ROXANA BANK is fitted with an exhaust gas boiler which theoretically produced 800 kg/hour from the exhaust gas, the exhaust gas reduced from 350° to 250°C. (Table 6.5 Heat Load).

Morton (1981:19) concludes that the recoverable power falls off very sharply as the exhaust temperature at the waste heat boiler is reduced, even if the gas flow rate is unaltered.

Wartsila engine manufacturers are aware of the substantial fuel savings using waste heat recovery from the diesel engine exhaust and high temperature cooling water systems. (Wartsila Diesel (a) 1985).

The basic heat recovery from the jacket water is 85 kilowatt per cylinder at 100% load and incorporating the high temperature section of the charge air cooler in the heat recovery circuit, the heat recovered is increased to 160 kilowatt per cylinder.

Therefore, elaborating on items (1), (2) and (3), the heat recovery prospects are most likely to come from:

(4) Exhaust gas after turbocharger - temperature differential 350° to 170°C.

(5) The high temperature section of the charge air cooler directly after the compressor 130° to 150°C.

(6) The jacket cooling water circuit with temperatures stated as 75°C, although Gallois (1981:9) has stated that temperatures of 130°C are possible.

A further requirement for total heat recovery is the use of non water cooled turbochargers. Using this type of turbocharger will increase the exhaust gas temperature by 10° to 15°C and hence the steam output from the exhaust gas boiler by approximately 8% (Anon 1988b:47). Gallois (1981:9) calculates an increase of 25°C after the turbocharger with no water cooling as the exhaust temperature from a medium speed diesel engine is higher than that of a slow speed diesel engine.

Exhaust gas boiler output rates will be dependent on the exhaust gas mass flow variations between large bore slow speed engines and the smaller bore medium speed engine.

It is appropriate at this stage to point out that there is little variation in the methods of heat recovery from the water circuits and exhaust gas. Engine operating parameters have changed very little with the temperature limitations placed on exhaust systems due to the lack of suitable ultra high temperature materials. It is noted that jacket water temperatures are increasing in order to increase the feed water temperature through heat transfer. An added advantage of increased cylinder water is that the chance of acid attack is reduced due to higher cylinder wall metal temperature.

As the jacket water differential temperature is still in the order of 9° to 10°C it follows that the heat load has changed very little. With the heat recovery systems being designed, a revised engine cooling system is required in order to maintain more accurate control of temperatures.

SULZER, WARTSILA (Wartsila Diesel (b) 1989:47) and ALFA LAVAL have pioneered central cooling systems and many owners are specifying these systems when ordering new vessels. Central cooling systems offer the avoidance of corrosion, fouling and clogging of heat exchangers as fresh water is used instead of sea water in the main cooling circuits. The fresh water is cooled in a central cooler where sea water is used as the cooling medium. As there is only one sea water cooler used, the unit can be placed close to the hull reducing the length of piping and number of valves (Krastins, not dated:3).

The unit is generally manufactured from titanium which is impervious to corrosion and has unique self healing properties when subjected to abrasions.

A popular system in use is the high temperature and low temperature circuits which are common about the central cooler and employ separate pumps to circulate fresh water through the high temperature circuit to the engine cylinders and fresh water distillers and the low temperature circuit to the lubricating oil cooler and the charge air cooler. There is mixing of the fluids prior to entry into the central cooler which is temperature controlled at an outlet temperature of 32°C. (Krastens: Not Dated:8).

Theron (1985:7) states that compared to conventional sea water cooled systems, centralised cooling systems are more expensive in terms of cooling surface areas, related to the central cooler as this unit is of titanium. However, all other heat exchangers, valves and piping can be of cheaper materials as they are in contact with fresh water only, where no corrosion will take place. With the high and low temperature circuits in use and the heat recovery process removing heat from high temperature system, optimum design of the central cooler can take place to accommodate a specified cooling load. Wartsila (Wartsila Diesel(b) 1989:40) gave developed a system utilising the low temperature fluid for heating the charge air at low engine load. The low temperature water can also be used to heat the suction air to the turbocharger if this unit is drawing air from outside the engine room. Conversions are also possible to existing cooling water systems in which all shell and tube heat exchangers are removed and replaced with plate heat exchangers. Alfa Laval Desalt suggest that improving thermal performance through a better cooling system is one of the fastest and least capital intensive measures open to a ship owner seeking to cut operating and maintenance costs (Anon 1988(e):15).

Substantial savings can also be achieved by controlling the sea water flow to suit the sea water temperature and/or heat load. Installations can consist of a number of smaller pumps to operate in parallel or two speed pumps with both with switching devices to suit (Theron 1988:10; Alfa Laval(b):6). A further system is a variable speed pump where the speed is adjusted by the use of a frequency converter.

Gibson (1989:11) writes that further advantages of this system is the reduction of high starting loads and power surges across the switchboard. System noise and vibration are also reduced. Variable frequency controllers are also very flexible and have the ability to handle various control signals such as air or electronic signals.

To ensure efficient heat recovery, selection of the correct type of heat exchanger for maximum effect must be specified. A further important factor is that the heat exchanger must be of a modular design and construction in order to enlarge or reduce the heat load if required.

Kern (1983:2 onwards) discusses the theory behind heat transfer and its application to heat exchanger design. The design is predominantly in favour of shell and tube type exchangers and the limitations are featured.

The theory behind heat exchanger design varies little between shell and tube type and plate heat exchangers.

The shell and tube heat exchanger can be manufactured in virtually any material whereas plate heat exchangers are limited to any material that can be pressed.

The initial costs for shell and tube heat exchangers and plate heat exchangers are similar due to the more construction material required, but plate heat exchangers are more capital consuming (Heat Exchanger Guide:4). Plate heat exchangers are much more compact than shell and tube types and require no servicing space. (Krastins, Not Dated:3).

The plate heat exchanger is designed primarily for liquid to liquid heat transfer, and in such service has outstanding performance. As an indication, a typical water to water overall heat transfer coefficient including a generous allowance of about 15% for fouling, in excess of 6 000 w/m^2K can be achieved at about 100 kPa. This is about three times the value obtained in a shell and tube heat exchanger. (Krastins - Not Dated:5).

To improve heat transfer coefficient, the fluid velocities must be of a magnitude so as to produce turbulent flow. Fluids are in the laminar or streamline Flow region when Reynolds numbers are below 2300 and fully turbulent when the Reynolds numbers exceed 20 000. Any values falling in between these numbers are in the transition region. (Kern 1983:36) exceed 20000. Any values falling in between these number are in the transition region. (Kern 1983:36).

Thomas and Hillis (1988:614) have carried out experiments to determine the effectiveness of various types of enhancements to tubes in order to promote better heat transfer rates and turbulent flow. However a price to pay for enhancing heat transfer surfaces is cost. Enhancement types include fin tubes, spiral agitators fitted inside tubes and spiral fluted tubes.

During experiments Cross (1979:87) has reported that plate heat exchangers induce high turbulence to fluids. In some units, turbulent flow can occur at Reynolds numbers as low as 10. In typical plate heat exchangers, hydraulic diameters are small, varying between 0,0048 and 0,011 m.

The most important factor affecting the fouling characteristics of a fluid with respect to a certain type of surface, is the fluid velocity and its resultant influence on shearing force and turbulence, laminar layer thickness and residence time close to the surface. (Alfa Laval (a) (undated):8) For more complex heat exchangers Pitts and Sissom (1975:343) state that those involving multiple tubes, several shell passes or crossflow configurations, determination of the average effective temperature difference is so difficult that the usual practice is to use a correction factor F_t.
Kern (1983:144,177) confirms this practice.

However, the Transon heat exchanger due to the nature of its design has the shell fluid in spiral counterflow (Transheat - undated) which significantly improves heat transfer conditions.

The choice for a low viscosity fluid to low viscosity fluid application the plate heat exchanger is found to be the most suitable. (Alfa Laval(a)(undated):11) Again the plate heat exchanger is the most suitable for medium and high viscosity fluids due to fluid velocity considerations.

When confirming specifications for a heat exchanger, the pressure drop must be evaluated to ensure correct sizing of the pump, pressure drop is the price paid for heat transfer. The heat exchanger should be designed in such a way that unproductive pressure drop is avoided to as high a degree as possible in other words, inlet and outlet return bend and valve losses must be minimised. The plate heat

exchanger can develop a much higher value per pass than any other heat exchanger type. The plate heat exchanger effectively uses pressure drop to produce heat transfer.
(Alfa Laval (a)undated:7)

The trial data (Appendix C.6,3) shows the operating parameters in a sea temperature of 4°C and from calculations shows that a volume flow of 14 m^3hour passing through the cylinder water cooler which results in a heat load of 580 kilowatts. It is obvious that the fresh water distiller is not on at this stage. This illustrates the under utilisation of the cylinder water cooler and the need for reduced sea water flow.

The provision of suitable waste heat recovery is probably justified in ships where electrical loadings for services other than propulsion plant auxiliaries are high for example, container ships, passenger ships which typically have large electricity and steam demands for accommodation and other services. (Anon, 1988(b):44) Wartsila have acknowledged that the demand for heat in fishing vessels I increasing. The application of heat recovery of any kind will refer to the large freezer trawler. (Wartsila(c) (undated):4) The influence of feed water temperature, pinch point and temperature after the exhaust gas boiler play an important part in heat exchanger size and a high feed temperature coupled with a small pinch point increases the size of the heat exchanger quite dramatically (A Special Survey:69). A calculation has been carried out showing the effect of reducing pinch point and increased heat recovery for the proposed high temperature heat recovery unit has on the increasing size of the heating surface area. (Appendix C.6.2a and 2b)

The costing of the conversion must be calculated as to the payback period and installation costs. Gallois (1981:10) had calculated in 1981 that for a 7500 kilowatt power and using a Pielstick PC4 engine, the saving will be 150 000 US dollars per annum, with an installation cost of 800 to 1000 US dollars per kilowatt for a recovered power of 1000 kilowatt or more and 1200 US dollars per kilowatt for 500 kilowatts recovered. The payback period is three to five years based on a fuel price of 175 US dollars per tonne. Fluctuations in fuel price are erratic with the present price at Durban port being 104 US dollars per tonne as quoted by Fairplay magazine (December 1989:10).

These fuel price figures play havoc with Gallois Calculations, but should calculations be required for local conversions, the exchange rate and inflation will play a large part in the costing of the project.

Love (Not Dated:9) has set guidelines to set target design objectives and how they are to be achieved. This has resulted in the design of the plant to suit the operating parameters as found in the trial data and present operating conditions as logged on the 09/10 November 1989.

Advancements in technology in heat exchangers particularly with shell and tube heat exchangers has been to improve the heat transfer coefficient, as this factor U_o has a direct influence on the heat exchanger surface area.

Thomas and Hillis (1988:615) recently tested a spirally fluted tube under OTEC conditions and obtained an overall heat transfer coefficient of 8 500 w/m^2K. There are however problems in the manufacture and sealing of this tube in the tube sheet. Studies have been carried out using ammonia in a closed circuit OTEC system. Thomas and Hillis (1988:608) used the refrigerant as the working fluid as the fluid properties were more suited to their experiments due to the temperature range. This system would be unsuitable for South African ships as the use of ammonia systems on ships is not permitted due to the toxicity of the fluid.

Miles (1988:18) confirms the continuous research and development into more efficient performance for plate heat exchangers. Units now feature thinner heat transfer surfaces which increases coefficients and longer thermal lengths produced by closer corrugations in the plates, allow closer temperature approaches down to 1K.

For the higher powered vessels (15 000 kW) on dedicated long voyages and short periods in port such as oil tankers and bulk carriers, e.g. a Mitsubishi D-Map Mark 2 system (Anon 1988(b):47) incorporating heat recovery from the air cooler, jacket water and exhaust gas with a steam turbine and gas turbine has been installed in the VLCC Tokyo Maru. The sea operations have been successful thus far (MER April 1988) and a further vessel is under construction with a refined Mitsubishi system to compete with the two European systems. The European systems are similar and incorporate a shaft generator to supplement the steam alternator. (Anon 1988(b):48). The Nissiki Maru is expected to consume only 48 tonnes of fuel per 24 hours at 14 knots. The fuel saving in this application is quite substantial considering the vessel is over the 300 000 tonne deadweight.

A further system suitable for evaluation is the thermal fluid system, using thermal oil which is suitable due to its high boiling point. This means that the oil can be circulated at high temperatures in its liquid state resulting in high heat transfer properties compared with water and steam.

A further system suitable for evaluation is the thermal fluid system, using thermal oil which is suitable due to its high boiling point. This means that the oil can be circulated at high temperatures in its liquid state resulting in high heat transfer properties compared with water and steam.

Thermal Fluid systems offer the following advantages:

(i) The system is open and therefore overpressure is not necessary;

(ii) Temperatures are possible up to 400°C;

(iii) There is no corrosion risk or freezing problems;

(iv) Low maintenance and operating costs.

A final system that has undergone testing on a small scale but theoretically shows good operating qualities is the organic fluid system in which certain refrigerants such as R11 (CCl_3F) and R113 (C_2Cl3F_3) have been analysed.

A disadvantage is that R11 and R113 require large mass flow rates to be effective, with the higher pumping power associated with the increased rates, due to high loading. Difficulties in the cycle in avoiding sudden temperature transitions from liquid to vapour are also experienced. A further disadvantage is the adverse publicity given to CFC's and it is unlikely that organic fluid systems will be found on fishing vessels.

Only R11 and R113 are suitable for these supercritical cycles. The efficiency of steam systems in practice (power output as a proportion of exhaust gas heat input) is only about 16% (MER March 82:13). Organic fluid cycles offer an increase in thermodynamic properties and higher efficiencies.

The increased efficiencies of these fluids is well documented (Morton 1981:17; Thomas and Hillis, 1988:608).

CHAPTER THREE

AN INTRODUCTION TO HEAT TRANSFER

3.1 Overview

The three methods of heat transfer namely conduction, convection and radiation are important in the field of heat recovery.

The most prevalent form of heat recovery envisaged on the ROXANA BANK is by conduction albeit turbulent or laminar flow.

The design of shell and tube heat exchangers along with the design for plate heat exchangers is investigated as these are the types of heat exchangers which will form part of this study.

Pressure drop is investigated as every heat exchanger will present a resistance to flow, the design of the unit having a direct influence on pressure drop and hence heat transfer.

3.2 Methods of Heat Transfer

3.2.1 Conduction

Conduction is the transfer of heat through a fixed material. The direction of heat flow will be at right angles to the wall if the wall surfaces are isothermal and the body homogeneous and isentropic.

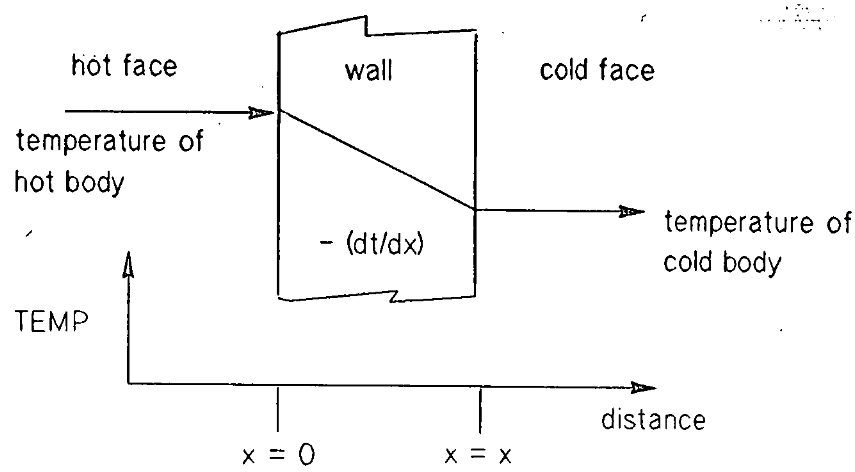


Figure 3.1 Heat Flow Through a Wall

Assuming that a source of heat exists on the left face of the wall and a receiver of heat exists on the right. It has been demonstrated by Kern (1983:2) that the flow of heat per hour is proportional to the change in temperature through the wall and the area of the wall "A".

If "t" is the temperature at any point in the wall and "x" is the thickness of the wall in the direction of heat flow, the quantity of heat flow dQ is given by Fouriers Law:

$$dQ = kA(-dt/dx) \quad \text{Kw} \qquad \text{-- (3-1)}$$

where the proportionality constant "k" is known as the thermal conductivity in w/mK and is peculiar to heat transfer. The minus sign in the term (-dt/dx) is added to satisfy the second law of thermodynamics which states: Thermal energy transfer resulting from a thermal gradient must be from a warmer to a colder region.

3.2.2 <u>Convection</u>

Convection is the transfer of heat between relatively hot and cold portions of a fluid by mixing. If the upstream temperature of the fluid is T_i and the downstream temperature is T_o, the heat transfer per unit time is:

$$Q = UA(T_i - T_o) \qquad \text{-- (3-2)}$$

this equation is known as Newtons law of cooling and the proportionality constant "U" is known as the heat transfer coefficient in w/m^2K.

3.2.3 <u>Radiation</u>

Radiation involves the transfer of radiant energy from a source to a receiver. While conduction and convection are proportional to a linear temperature difference, experiments have shown that radiant heat transfer is proportional to the fourth of the absolute temperature.

From the Stefan - Boltzmann Law:

$$Q = \epsilon\sigma AT^4 \qquad \text{-- (3-3)}$$

which is based on the second law of thermodynamics where σ is a dimensional constant of value 5,6697 * 10^8 w/m^2K and ε is the emissivity, a factor peculiar to radiation and ranges from zero to one depending on the type of surface.

3.3 Thermal Conductivity

The thermal conductivities of solids have a range of numerical values depending on whether the solid is a relatively good conductor of heat such as a metal or a poor conductor such as asbestos. (Appendix D.1.2, Appendix D.1.3). The thermal conductivity of the solid phase of the metal is primarily dependent on temperature. Pitts and Sissom (1977:2) state that in general "k" for a pure metal decreases with temperature increase, while alloying elements tend to reverse this trend. Thermal conductivities of most liquids decrease with increasing temperature (Appendix D.1.2) with the exception of water which exhibits increasing "k" up to about 150°C and decreasing thereafter. Water has the highest thermal conductivity of all liquids with the exception of liquid metals.

The thermal conductivity of a gas increases with increasing temperatures but essentially independent of pressure for pressures close to atmosphere. (Appendix D.1.5.)

3.4 The Heat Transfer Coefficient

(Referring to a single tube or a double pipe heat exchanger)

Consider a pipe wall with forced convection of different magnitudes on both sides of the pipe as shown in figure 3.2: Heat Flow Through a Wall.

Heat flowing on the inside is transferred to a cold flowing liquid on the outside. The heat transfer can be calculated from the sensible heat exchange in either the hot or the cold fluid.

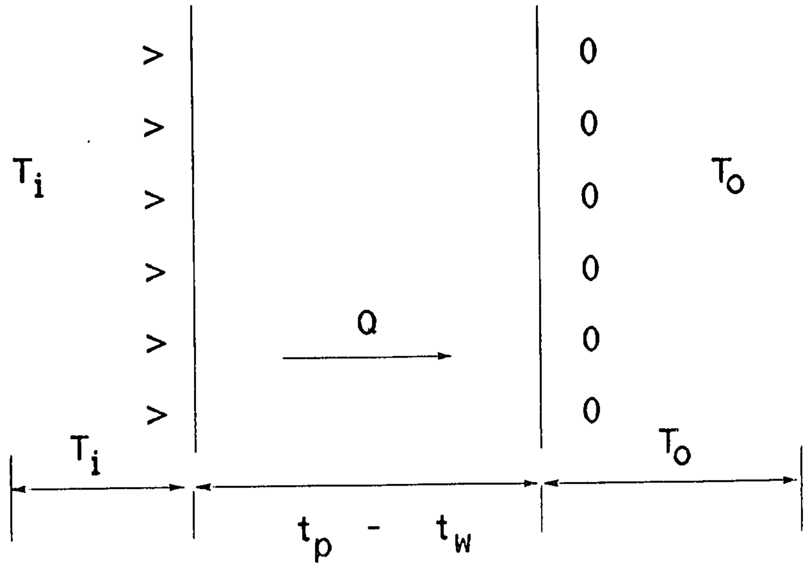


Figure 3.2 Heat Transfer Through a Wall

Designating the inside resistance R_i, the outside resistance R_o, the inside and outside pipe wall by t_p and t_w and applying an expression for the steady state:

$$Q = A_i(T_i - T_o)R_i = A_o(tw - t0)R_o \qquad \text{-- (3-4)}$$

where T_i is the temperature of the hot fluid on the inside and TO the temperature of the cold fluid on the outside. Replacing the resistances by their reciprocals h_i and ho:

$$Q = h_iA_i\Delta t_i = h_OAO\Delta t_O \qquad \text{-- (3-4a)}$$

where h_i and h_O are called the film coefficients in w/m^2K. The film coefficient is a measure of the heat flow for unit surface area and unit temperature difference.

As h_i has been determined for the inside diameter of the tube, this value must be corrected for the tube wall thickness and referred to the outer diameter of the tube.

To find the overall coefficient before taking into account the dirt factors:

$$U_c = \frac{1}{r_i/(rh_o) + [r_i \ln(r_ir_o)]/k + l/h_o} \qquad \text{-- (3-5)}$$

The greater difference between the hot and cold temperatures (temperature driving force), the easier the heat transfer will be. The resistance of the material separating the two fluids is another consideration, but because thermal conductivity in metals is good, wall resistance is one of the less serious problems.

Fouling, which is chemical, corrosive or organic buildup of deposits seriously impedes heat transfer and must be removed to keep performance from degrading. With the determination of the clean coefficient the dirt factors must be considered to find the design coefficient. These dirt factors (or fouling factors) Rd may be considered very thin for dirt but may be appreciable thick for scale. The dirt factors have been determined by evaluation of actual conditions in a large number of applications.

A major resistance to heat transfer is the boundary film of fluid separating the wall from the bulk of the fluid stream. Turbulence and enhancement can reduce this resistance.

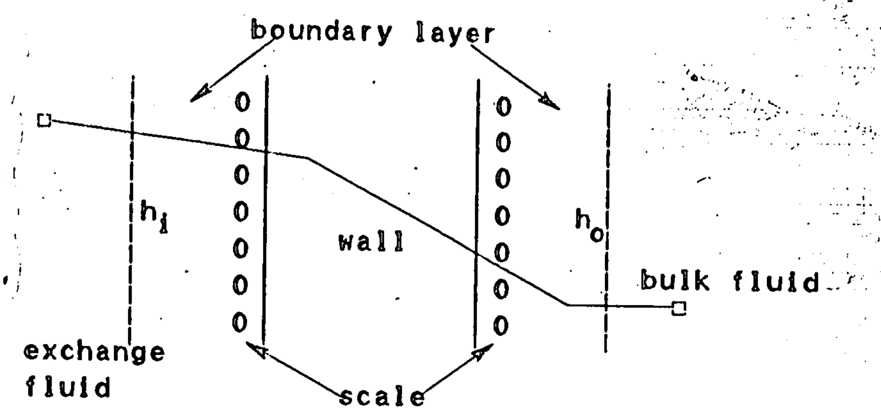


Figure 3.3 Position of Fouling and Dirt Factors

Combining the clean coefficient and dirt factors, the design coefficient may be determined and from this equipment may be designed.

Thus $R_d = R_{di} + R_{do}$ -- (3-6a)

and $U_d = 1/U_c + R_d$ -- (3-6b)

and the Fourier equation $Q = U_d A \Delta_t$ allows the surface area to be calculated.

When the fluid is viscous or the pipe wall temperature varies considerably from the bulk temperature, a viscosity correction (ϕ) must be taken into account.

Thus :

$\phi = (\mu/\mu_w)E^{0,14}$ and for corrected coefficients

$h_o = (h_o/o_a)o_a$ where $\phi \neq 1$ -- (3-6c)

$h_{io} = (h_{io}/o_o)o_p$

From which the substitution can be made to evaluate U_c.

3.5 Principals of Convection

The subject of convection heat transfer requires an energy balance along with an analysis of the fluid dynamics of the problems concerned.

An important factor both on convection and fluid flow is viscosity. In order to evaluate this property by fluid dynamics, two assumptions are required:

3.5.1 Where a solid/liquid interface exists, there is no slip between the solid and the liquid.

3.5.2 Newtons rule: Shear stress is proportional to the rate of shear in the direction perpendicular to motion. (Kern 1983:27)

The rate of shear is proportional to the velocity gradient as shown in Figure 3.4. Fluid Shear.

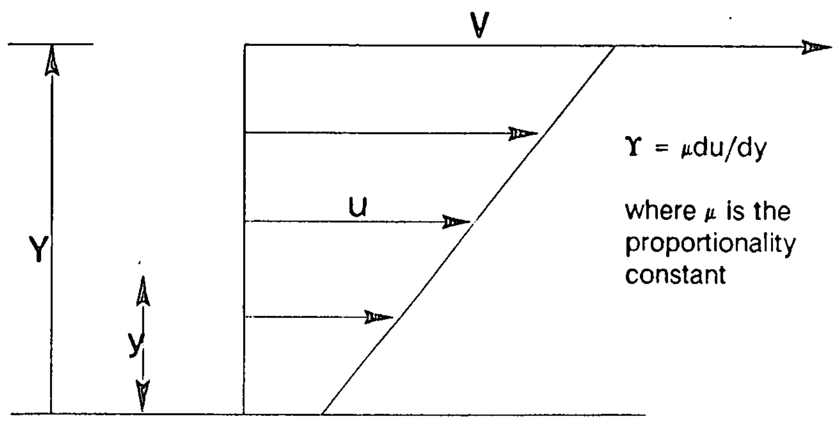


Figure 3.4 Fluid Shear

The rate of fluid shear is given by du/dv or by V/Y giving $\gamma = \mu V/Y$ where μ is called the viscosity when V and Y have unit values. μ is called the dynamic viscosity with typical units of Ns/m^2 or the mass based units kg/m s.

3.6 Heat Transfer Between Fluids (Forced Convection)

3.6.1 Streamline (Laminar) Flow

If the linear velocity of the liquid is decreased below some threshold value the nature of the flow changes, and the fluid particles flow in lines along the axis of the pipe as shown in Figure 3.5 Streamline Flow.

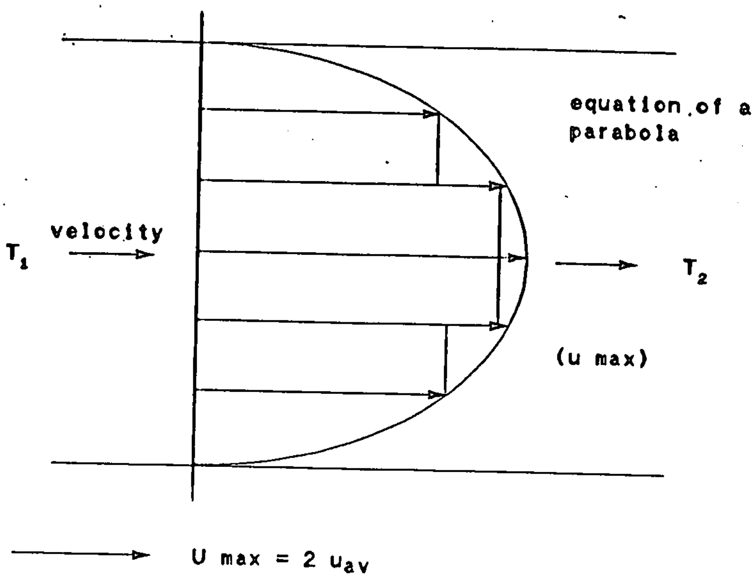


Figure 3.5 Streamline Flow

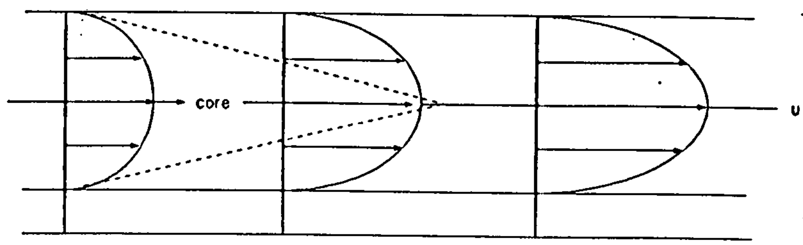
Reynolds observed that the type of flow assumed by a liquid flowing in a tube was influenced by velocity, density, viscosity and the tube diameter. When related as the quotient of vd/ called the Reynolds number (dimensionless) he found that streamline flow usually existed below 2300 and always below 2100.

By definition, the transfer of heat by convection proceeds mainly by mixing and while this requirement appears to be satisfied by turbulent flow, it is not fulfilled by streamline flow.

Streamline flow is a form of conduction and the speed at which heat is transferred to or from a liquid is considerably less for streamline flow.

Considering the theory of increasing thermal boundary layer on a flat plate and applying this theory to tube or duct flow, the boundary layer thickness in a pipe is physically restricted to the radius of the pipe.

Figure 3.6 Developing boundary layer and velocity - illustrates the successive stages of development of the boundary layer in the entrance region of a circular tube.



$u = 1(4_\mu)[(dp/dx)(R^2 \times r^2)]$

Figure 3.6 Developing Boundary Layer and Velocity

At the tubes entrance slug flow or uniform flow at the free stream flow exists. As the fluid moves down the tube, the shear between the fluid and the wall and between the adjacent fluid particles retards the motion until it is fully developed at station 3. From this point on the velocity profile remains unchanged.

Although heat transfer in laminar flow is not too common because the transfer rate is lower than encountered in turbulent flow, it is sometimes desirable due to the low pumping power required. In pure laminar flow the heat transfer mechanism is conduction.

The length required for the velocity profile to become invariant with axial position is known as the entry length (x_e).

Pitts and Sissom (1977:136) have found that the Langhaar equation approximates the entry length:

$$x_e = (0{,}05)R_eD \qquad \text{-- (3-7)}$$

Pitts and Sissom (1977:138) present the Hansen empirical relation for fully developed laminar flow in tubes at constant temperature and parabolic inlet velocity:

$$Nu_d = 3{,}656 + \{0{,}0668[(D/L)RePr]\}/\{1+0{,}04[(D/L)RePr]^{2/3}\} \qquad \text{-- (3-8)}$$

where $Nu = (hd)/k$ -- (3-8a)

$Re = (\rho vd)/\mu$ -- (3-8b)

$Pr = (c\mu)/k$ -- (3-8c)

For oils or fluids in which viscosity varies considerably with temperature, the second term must be corrected by:

$$\phi = (\mu/\mu_w)0{,}14 \qquad \text{-- (3-8d)}$$

The heat transfer coefficient calculated from this relationship is the average value taken over the entire length of the tube. Kern (1983:103) discusses a simpler empirical formula which has been proposed by Sieder and Tate who made a correlation by heating and cooling a number of fluids when $Re<2100$.

$$Nu = 1{,}86[RePr(D/L)]1/3(\mu/\mu_w)^{0{,}14} \qquad \text{-- (3-9)}$$

In this formula the average heat transfer coefficient is based on the arithmetic average of the inlet and outlet temperature difference and all the fluid properties are evaluated at mean bulk temperature.

During the design and evaluation of the feed system and heat recovery units, the phenomena of laminar flow will become apparent, particularly the main engine lubricating oil purifier preheater.

3.6.2 Turbulent Flow

When liquid flows in a horizontal pipe it may flow with a random eddying motion known as turbulent flow as shown in Figure 3.7 Velocity Profile of Turbulent Flow.

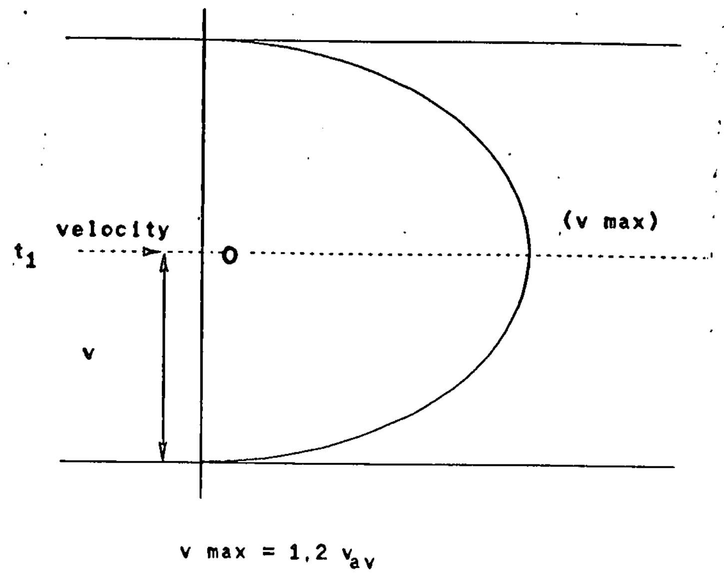


Figure 3.7 Velocity Profile of Turbulent Flow

Initially the boundary layer development is laminar, but at some critical distance from the leading edge, depending on the flow field and fluid properties, small disturbances in the flow become amplified, and a transition process takes place until the flow become turbulent. The turbulent flow region may be pictured as a random churning action with chunks of fluid moving to and fro in all direction.

Pitts and Sissom (1977:170) recommend the relation developed by Dittus and Boelter:

$Nu = 0{,}026Re^{0{,}8}Pr^{0{,}3}$ for cooling -- (3-10a)

$Nu = 0{,}023Re^{0{,}8}Pr^{0{,}4}$ for heating -- (3-10b)

Taking into account the property variations suggested by Sieder and Tate in Kern (1983:103) recommends the following relation:

$$Nu = 0{,}027\ Re^{0{,}8}Pr^{1/3}\ (\mu/\mu_w)_{0{,}14} \qquad \text{-- (3-11)}$$

The above equations were obtained for tubes. They will be used indiscriminately for pipes. Pipes are rougher than tubes and hence produce more turbulence for equal Reynolds numbers.

Gnielinski (Kern 1983:103) has developed a more accurate expression for flow in smooth tubes:

$$Nu = \frac{\{f[Re-1000]Pr[1+(D/L)^{2/3}]\}}{\{8+101{,}6(f/8)^{0{,}55}(Pr^{2/3}-1)\}} \qquad \text{-- (3-12)}$$

where $f = (1{,}821ogRe-6{,}14)^{-2}$ -- (3-12b)

for a smooth tube. This equation is valid for Re>2300.

3.7 Heat Exchangers - Methods of Design

The primary objective in the thermal design of heat exchangers is to determine the necessary surface area required to transfer heat at a given rate for fluid temperatures and flow rates.

There are two methods of calculating heat exchanger surface areas.

3.7.1 The Log Mean Temperature Difference (LMTD) Method From the Fourier equation

$$Q_d = UA\Delta_t$$

U_d is the design coefficient and is found from equation (3-5) and (3-6a) and b): from which h_o and h_i are calculated from the Nusselt Number.

$$Nu = (hd)/k \text{ and}$$

$$Nu = (h_oDe)/k = 0{,}36Re^{0{,}55}Pr^{1/3}(\mu/\mu_w)^{0{,}14} \qquad \text{-- (3-13a)}$$

$$Nu = (h_iD)/k = 0{,}027Re^{0{,}8}Pr^{1/3}(\mu/\mu_w)^{0{,}14} \qquad \text{-- (3-13b)}$$

where $Pr = (c\mu/k)$ and $Re>2000$.

Log Mean Temperature Difference

In using this method, both inlet and outlet temperature of the primary and secondary fluids must be known as the log mean temperature difference (LMTD) may be calculated as follows:

$$\Delta T_1 = T_{hi} - t_{co}$$

$$\Delta T_2 = T_{ho} - t_{ci}$$

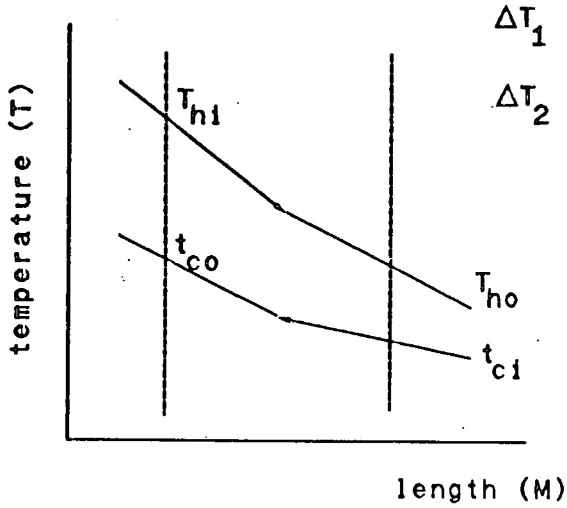


Figure 3.8 Log Mean Temperature Difference

It has been proved that LMTD = $(\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$ -- (3-14)

To obtain the heat load five of the following data are required:

(1) Primary fluid inlet and outlet temperatures

(2) Secondary fluid inlet and outlet temperatures

(3) Primary fluid volume flow

(4) Secondary fluid volume flow

The heat load can be calculated from:

$$Q = MC_p\Delta_t \quad \text{-- (3-15)}$$

and the unknown data is calculated from the heat balance equation.

Should the heat exchanger have multiple tubes, several shell passes or crossflow, a correction factor F_t will modify the Fourier equation to:

$$Q = U_d A F_t (LMTD) \qquad \text{-- (3-15A)}$$

3.7.2 Heat Exchanger Effectiveness (NTU Method)

If more than one of the inlet or outlet temperatures of the heat exchanger are unknown, the LMTD method is unwieldy, requiring a trial and error iterative approach.

Another approach introduced a definition of heat exchanger effectiveness.

$$E = \frac{\text{actual heat transfer}}{\text{max heat transfer possible}} \qquad E = \frac{Q_{(actual)}}{Q_{(max)}}$$

Where the maximum possible heat transfer is that which would result if one fluid underwent a temperature change equal to the maximum temperature difference available - the temperature of the entering hot fluid minus the temperature of the entering cold fluid This method uses the effectiveness E to eliminate the unknown discharge temperature and gives a solution for effectiveness in terms of other known parameters.

Letting $C = MC_p$

$$Q_{(actual)} = C_h(T_{hi} - T_{ho}) - C_c(T_{co} - T_{ci}) \qquad \text{-- (3-16)}$$

which indicates that the energy given up by the hot fluid is gained by the cold fluid. The maximum temperature difference occurs when the fluid of the smaller C undergoes the maximum temperature difference available.

$$Q_{(max)} = C_{min}(T_{hi} - T_{ci}) \qquad \text{-- (3-16a)}$$

This transfer would be attained in a counterflow heat exchanger of infinite area. Combining E and Qmax, the basic equation to determine heat transfer is:

$$Q_{actual} = EC_{min}(T_{hi}-T_{ci}) \quad \text{-- (3-16b)}$$

It has been proved that the effectiveness for a parallelflow double pipe heat exchanger is :

$$E = \{1 - \exp[-N(1+C)]\}/(1+C) \quad \text{-- (3-17)}$$

and for a counterflow exchanger:

$$E = \{1 - \exp[-N(1-C)]\}/\{1 - C\exp[-N(1-C)]\} \quad \text{-- (3-18)}$$

(Heat Transfer:247)

where $N = (UA)/C_{min}$ and $C = C_{min}/C_{max}$ -- (3-18a)
-- (3-18b)

The grouping of the terms UA/C_{min} is called the number of transfer units or NTU since it is indicative of the size of the heat exchanger, and is also referred to as the thermal length.

Generally speaking, a shell and tube type heat exchanger applied to a water/water duty will achieve an effectiveness of about 0,5 HTU per pass at moderate pressure drops.
Compact heat exchangers - particularly the plate heat exchanger can have thermally long channels such that the effectiveness can be as high a 4 HTU per pass.

3.8 Heat Exchangers

The choice of heat exchanger type for a particular duty depends on a large number of factors among which the most important are space, fluid properties, temperature and pressure programs, maintenance and cost.

The performance of a heat exchanger may be expressed in terms of the θ value (number of HTU) of which the heat exchanger is capable of.

ϕ may also be defined as the ratio of the temperature change of one fluid to the mean temperature difference between the two fluids.

Thus $\phi = (t_i - t_o)/V\Delta_{tm}$ -- (3-19)

From the basic equation:

$$\phi = kA\Delta t_m = V\rho C_p(t_i . t_o) = MC_p(t_i - t_o) \qquad \text{-- (3-20)}$$

It can also be seen that ϕ can be defined as:

$$\phi = (kA)MC_p \qquad \text{-- (3-21)}$$

(in these equations $V\rho C_p$ MC_p and $[t_i - t_o]$ must refer to the same fluid).

The following types of heat exchangers are considered in this study only:

3.8.1 Shell and tube heat exchangers

Single and multipass (primary and secondary sides) including double pipe heat exchangers.

3.8.2 Gasketed plate heat exchangers (PHE)

Before selecting a heat exchanger for a given duty, the following points must be considered:

(1) Materials of construction

(2) Pressure and temperature

(3) Performance parameters - temperature program, flow rates and pressure drops

(4) Fouling tendencies

(5) Inspection, cleaning, extension and repair possibilities

(6) Types and phases of fluids

(7) Overall economy

3.8.1 The Shell and Tube Heat Exchanger (THE)

The THE can be manufactured from virtually any material which may be required for corrosion resistance. The simplest form THE is the double pipe exchanger which consists of two concentric pipes where one fluid flows in the inner pipe and the other in the annulus.

Fluids may be parallel or counterflow.

The double pipe exchanger is extremely useful as it can be assembled in any workshop as piping, connecting Tees and return heads are all standard and provide inexpensive heat transfer surface.

The principal disadvantage of double pipe heat exchangers lies in the small amount of heat transfer surface area and that with all the connections is prone to leakage. A further disadvantage is that as the pipe sizes are standard a problem could occur in maintaining mass flow rates and fluid velocities resulting in laminar flows and high or low pressure drops. Though experience Kern (1983:103) has found that the time and expense involved in dismantling this type of equipment for cleaning is prohibitive compared with other types of equipment.

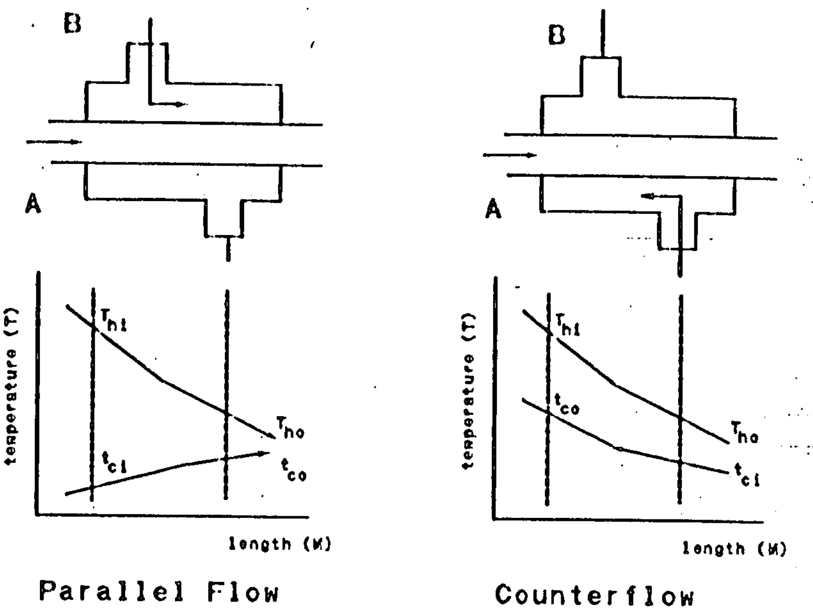


Figure 3.9 Parallel and counterflow on double pipe exchangers

The shell and tubes exchanger consists of:

(i) Shell - manufactured from steel pipe with nominal IPS diameters for up to 300 mm (12"). The standard wall thickness for shells from 300 to 400 mm (12 to 24") is 9,5 mm ($^3/_8$") which is satisfactory for pressures up to 2000 kPa. Shells above 600 mm (24") are fabricated from rolled steel plate.

(ii) Tubes, pitch and tube sheets: Tubes are available in a variety of metals including steel, copper, 70/30 copper nickel, Incaloy 825. They are available in various thicknesses known as the Birmingham Wire Gauge (BWG). Common layouts for tube in tube sheets which are standard are:

(a) Square pitch

(b) Triangular pitch

(c) Square pitch rotated

(d) Triangular pitch with cleaning lanes

(Source: Kern 1983:845)

The pitch P_t is precalculated and along with the tube counts sizes may be determined from the tables issued by TEMA. In a crossflow application the least pressure drop will be obtained for standard square pitch, but more turbulence will be obtained by using square pitch rotated or triangular pitch, the triangular pitch being the most efficient because fluid flowing between adjacent tubes impinges directly on to the succeeding row.

In order to obtain higher heat transfer coefficients and maintain a liquid in a state of turbulence, baffles are used in shell applications. These baffles will cause the fluid to flow through the shell at right angles to the axes of the tubes.

The drawbacks of this system is that there are considerable areas of stagnation and regions of low velocity occur in the shell resulting in a loss of efficiency. The fitting of baffles causes appreciable pressure drops but increases turbulence even for small quantities of liquid.

Kern (1983:147) shows how the following formula illustrates the effect the number of baffles "N" will influence pressure drop:

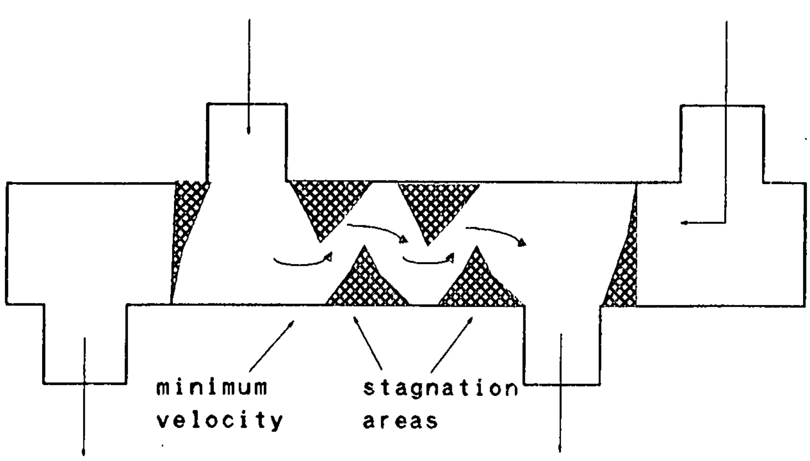


Figure 3.10 Single Pass THE Showing Efficiency Losses

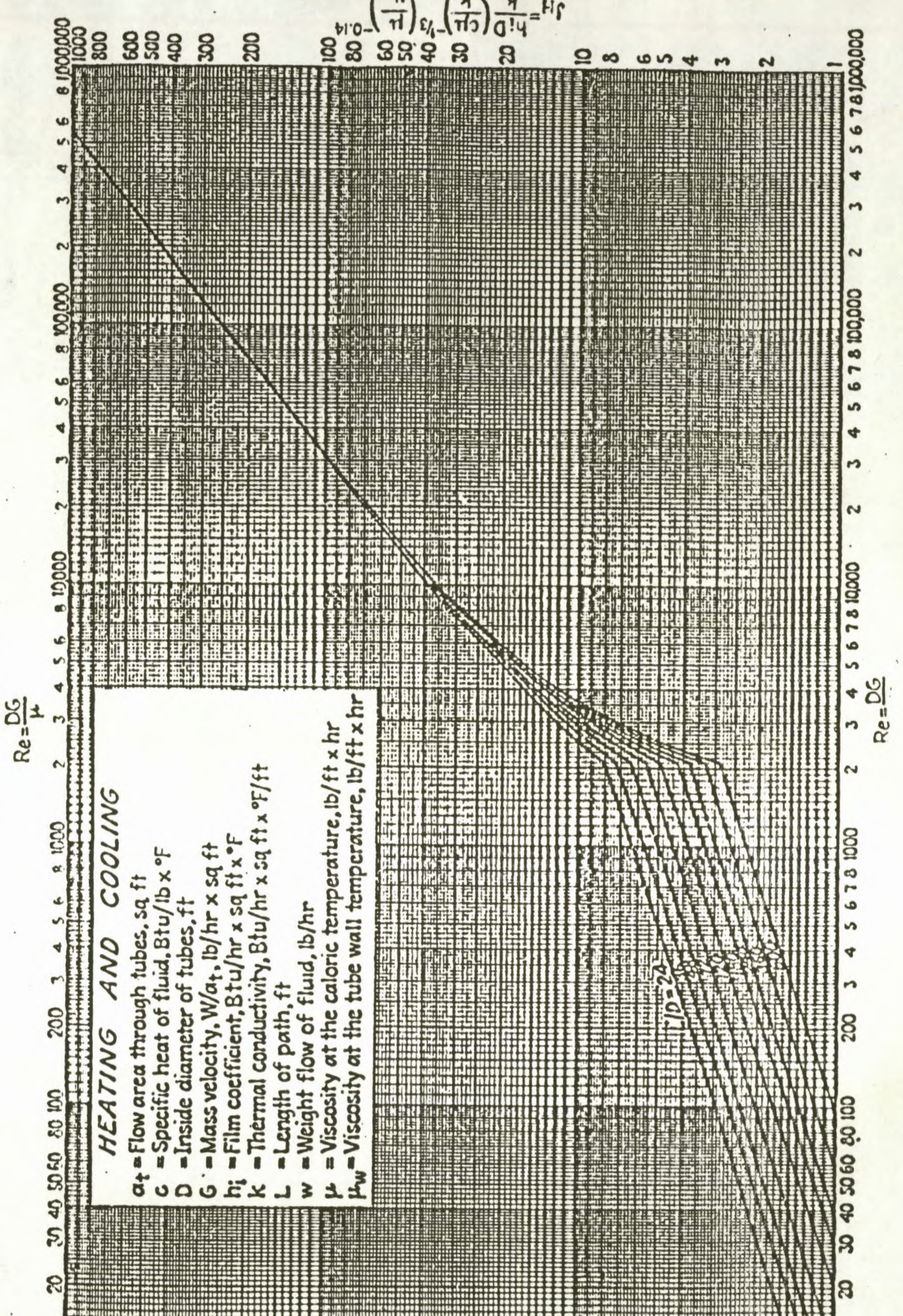


Figure 3.11 Heat Transfer Factor jH

The shell side coefficient is affected by the type of pitch, tube size, clearance and fluid flow characteristics in addition to the baffle spacing. Furthermore there is no true flow area by which the shell side mass velocity can be calculated since the flow area varies across the diameter of the tube bundle with the different number of tube clearances in each longitudinal row of tubes.

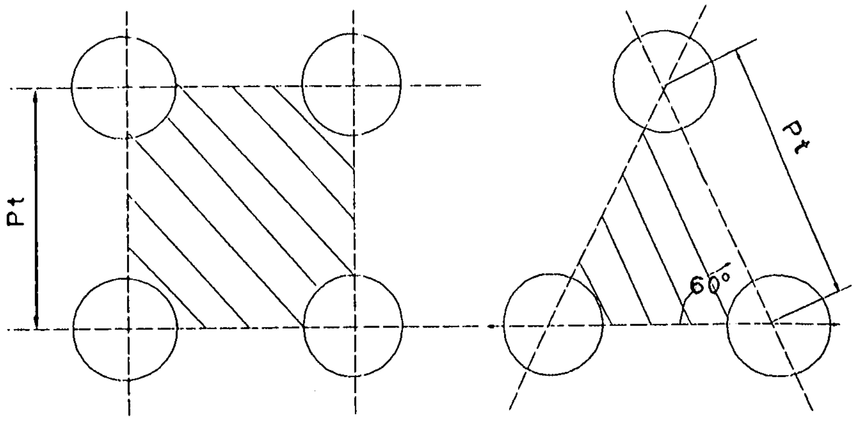
Figure 3.11 Heat Transfer Factor jH is a correlation of industrial data which gives satisfactory results for hydrocarbons, organic substances and water in the form of jH which is known as the heat transfer factor.

$$jH = [(hd)/k][c\mu)/k]^{1/3}(\mu/\mu_w)^{0,4} \text{ vs } [(DG)/\mu] \qquad \text{-- (3-23)}$$

However, for the values of the Re 2000 to 1000000 the data are closely represented by the equation: (Kern:137)

$$(h_oD_e)/k = 0,36[(D_eG_s)/\mu]^{0,55}(C\mu)/k^{1/3}(\mu/\mu_w)0,14 \qquad \text{-- (3.24)}$$

$$\text{where } De = \frac{4 * \text{free area}}{\text{wetted perimeter}} \qquad \text{-- (3-25)}$$



For Square pitch $de = [4[P_{12} -(\pi d_{o2}4)]/\pi d_o$

For Triangular pitch $de = \frac{\{[0.5P_t(0.86P_t-[0.5\pi d_{o2}]/4)\}}{\pi/(2d_o)}$ -- (3.25a) -- (3.25b)

Figure 3.12 Square and Triangular Pitch

In multipass THE's (on the tube side) the fluid in the tubes is relative to the shell fluid and having inlet and exit nozzles at the same end will have 50% in counterflow. Since greater temperature differences have been found when the process streams are in counterflow than in parallel flow, the LMTD for the two flows will be different.

It is therefore necessary to develop a new equation to calculate the effective or true temperature difference. The method used by Kern (1983:140) is a modification of the method used by Underwood and presented in its final form by Nagle and Bowman, Mueller and Nagle.

Equating $Q = U_dA\Delta_t$: $mc_p(T_i - T_o) = MC_p(t_i - t_o)$ and calling the fractional ratio of the true temperature difference to the LMTD, F_t,

The Fourier equation can be written $Q = U_dAF2t(LMTD)$ where

$$Ft = \frac{\sqrt{(R^2+1)}\ \ln (1-S)/(1-RS)}{(R-1)\ \ln \dfrac{[2-S(R+1-\sqrt{R^2+1})]}{[2-S(R+1+\sqrt{(R^2+1)})]}} \qquad \text{-- (3-26)}$$

where R is the ratio of the temperature range of the hot fluid to the temperature range of the cold fluid.

in other words $R = (T_i - T_o)/(t_i - t_o)$ -- (3.26a)

and S is the ratio of the temperature range of the cold fluid to the difference between the hot fluid inlet temperature and the cold fluid outlet temperature.

in other words $S = (t_i - t_o)/(T_i - t_o)$ -- (3-26b)

F_t will always be less than 1,0 due to the effect of counterflow and parallel flow occurring in the heat exchanger. Also it is advised not to use a 1-2 exchanger when the correction factor F_t is computed to be less than 0,75.

When a tube bundle contains four or more passes with equal surface area in each pass, it is known as a 2-4 exchanger. When designing and specifying a 2-4 exchanger, great care must be taken that a cross does not occur, resulting in the shell fluid which is being cooled is actually heated.

A 2-4 exchanger may be used when the process temperature gives an F_t value of less than 0,75 for a 1-2 exchanger. The correct number of passes will be determined by the value F_t being 0,85 to 0,9 which is the lower limit.

There are instances when the temperature cross is so great that only true counterflow can be used. A true shell and tube counterflow heat exchanger is manufactured and marketed in South Africa by Transheat (Pty) Ltd of Pinetown in which segment plates are arranged to impart a spiral counterflow pattern. This feature ensures significantly improved heat transfer conditions as the arrangement imparts a flow pattern to the fluid as if it were in a pipe, thus increasing the Reynolds and Nusselt numbers, imparting high velocities and eliminating dead spots. (Appendix A.4.2a)

These segments are adjustable in pitch and will therefore adjust the Reynolds number with the attendant increase or decrease in pressure drop. The heat exchangers are modular, easily maintained and due to the unique header design, the system is extendable. There is complete fluid separation with fail safe operation. The important limitation of the 1-2 exchanger lies in their inherent inability to provide effective heat recovery. When a temperature occurs in a 1-2 exchanger, the value of F_t drops sharply. Temperature cross occurs if $t_{co} > T_{ho}$ then $(t_{co} - T_{ho})$ is the temperature cross.

Assume conditions in which the shell fluid is reduced from 200° to 140°C, while the tube fluid rises from 80° to 160°C. From figure 3.13 Temperature Relations in a 1-2 Exchanger, it can be seen that all the heat from 140° to 80°C is lost whatever the configuration because of the close approach required between the tube fluid at the end of the parallel pass and shell fluid outlet.

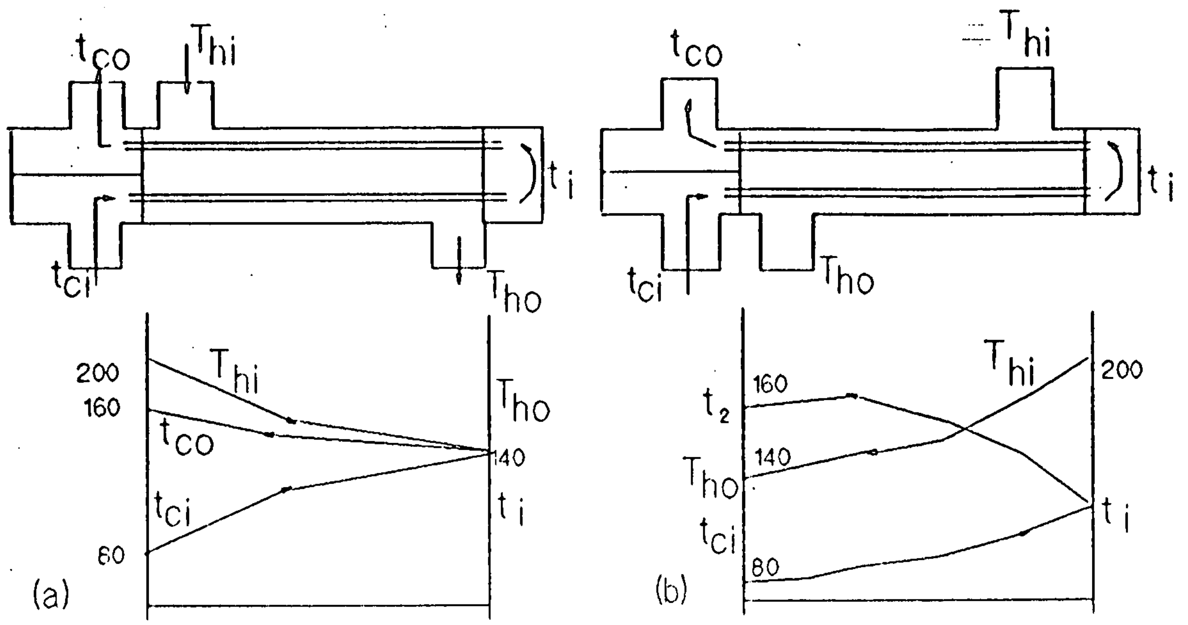


Figure 3.13 Temperature Relations in a 1-2 Exchanger

From figure 3.13b it can be seen that across exists so that the fluid leaving the shell at 140°C is forced to pass over the tubes carrying heated cold fluid. Thus the shell fluid may be cooled at some point to a temperature lower than its outlet and the tube fluid may be heated to a temperature above its outlet. When the two fluids are near their outlets, the shell fluid being cooled is actually heated and the tube fluid is actually cooled. This phenomena is called reheating. To prevent reheating a 2-4 exchanger may be used in which the longitudinal baffle reduces the effect as shown in figure 3.14 Temperature Relations in a 2-4 Exchanger.

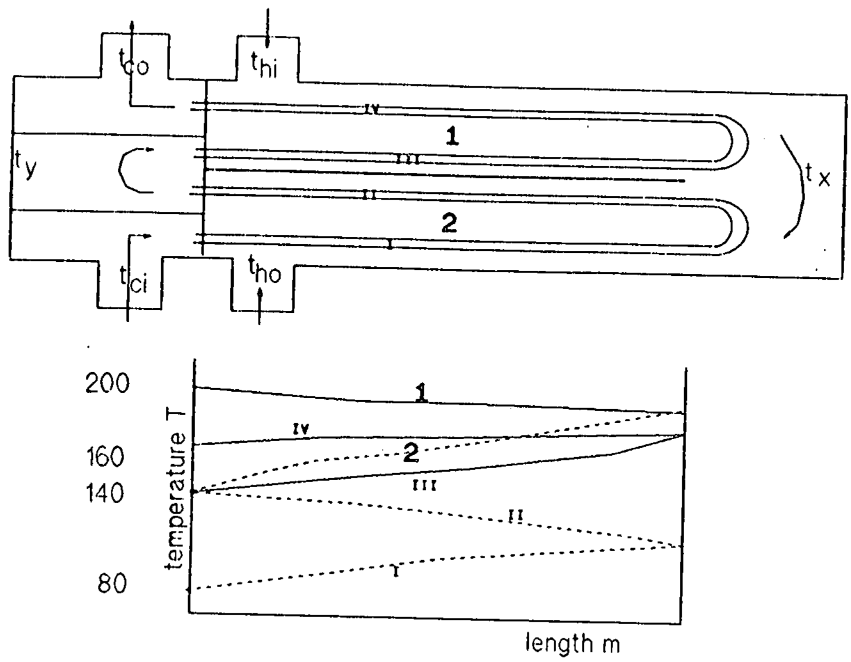


Figure 3.14 Temperature Relations in a 2-4 Exchanger

3.8.2 The Gasketted Plate Heat Exchanger (PHE)

The manufacture of PHE's is more capital consuming than for the THE's and relatively expensive materials are used in their construction. The cost per unit area for compact types of heat exchanger are higher than for THE's but the higher efficiency resulting in smaller surface area requirements may more than compensate for this disadvantage.

Compact heat exchangers are constructed of thinner materials and sudden changes in cross section are generally absent, although these smaller dimensions result in lower operating pressures and temperatures, the vibrations, fatigue and thermal effects found in THE's are absent.

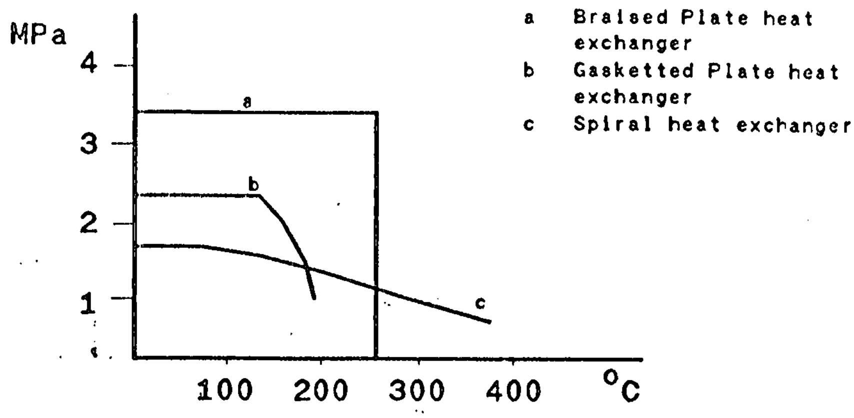


Figure 3.15 Operating Temperatures and Pressures for Compact Heat Exchangers

(Source: Heat Exchanger Guide:5)

A gasketted plate heat exchanger consists of the following main parts: (fig3:16)

(1) Frame plate to which all piping is connected

(2) Upper carrier bar

(3) Lower guide bar

(4) Support column

(5) Pressure plate

(6) Tie bolts

(7) Plate stack including gaskets

As the PHE can have thermally "long" channels they are therefore more likely to be single pass than for a THE for the same duty. Full counterflow with neither crossflow nor parallel flow to reduce the effective value of the LMTD is obtained. A high degree of temperature crossover can be considered possible.

The advantage of a gasketted plate heat exchanger are:

3.8.2.1 High heat transfer coefficient. The plate patterns produce very high turbulence and typical figure for the PHE including margins for fouling on water/water applications are 4 000 to 6 000 w/m^{2}K. The smaller surface area for the PHE for the same application is therefore apparent.

3.8.2.2 The risks of leakage between liquids is eliminated as each plate is pressed in one place and the gaskets are designed in such a way that mixture of the liquids is impossible.

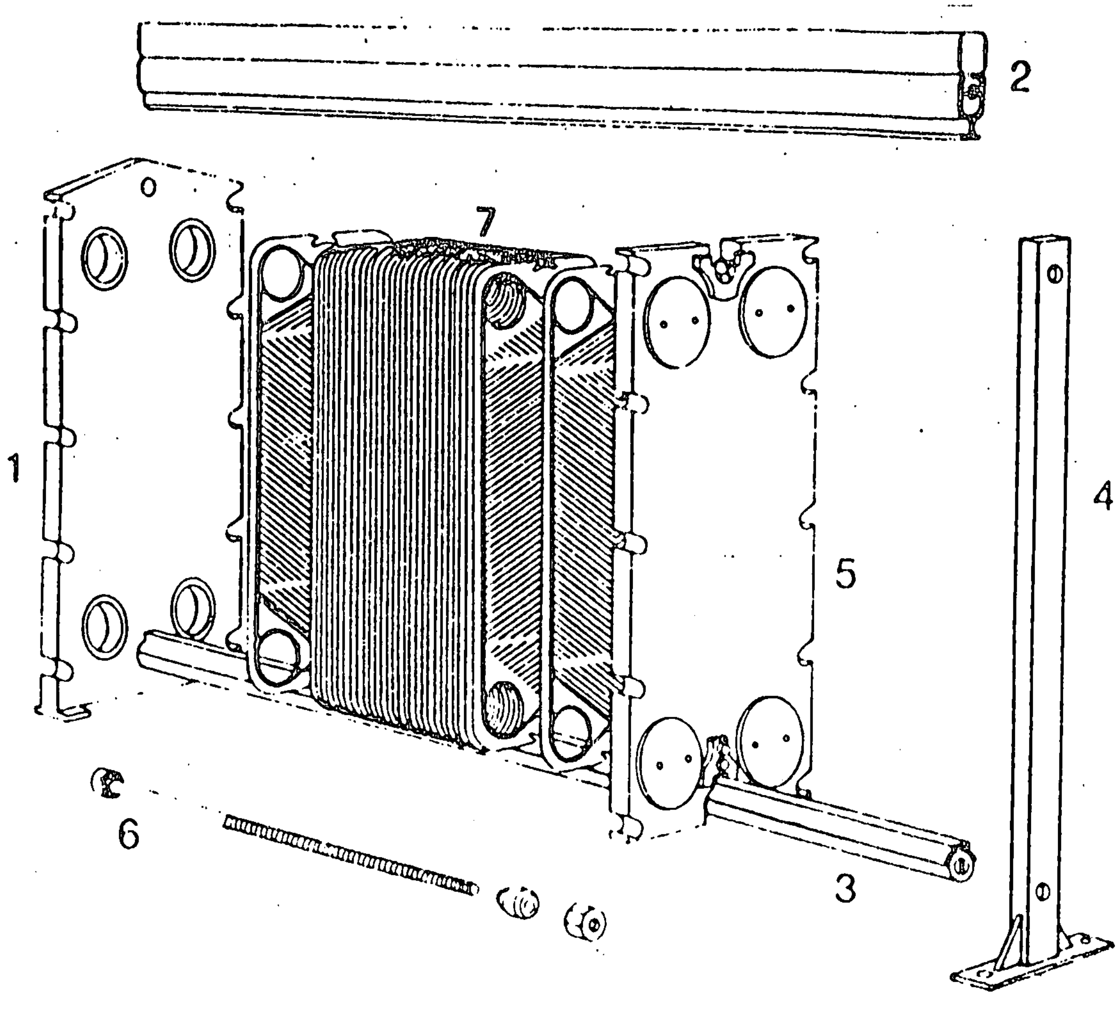


Figure 3.16 Exploded view of a Gasketted Plate Heat Exchanger

(Source: Plate Heat Exchangers:42)

3.8.2.3 The exchanger can be expanded for a greater heat load simply by adding more pairs of plates.

3.8.2.4 The PHE is easily cleaned and requires less space than a THE.

In order to calculate thermal lengths for plate heat exchangers, the 0 value must be calculated for a set of plate elements, therefore:

$$\phi = (t_i - t_o)/\Delta_{tm} = k2A/MC_p \qquad \text{-- (3-27)}$$

In this equation A is the heat transfer area per plate in m^2. (2A - area per channel bounded by a pair of plates).

The total thermal length ϕ of the PHE is the sum of all the θ values of channels connected in series and is thus limited to multiples of the uniform θ value of a given channel and overdimensioning can result.

3.9 Crossflow Heat Exchangers

Up to this point only parallel and counterflow heat exchangers have been discussed, with the difficulties experienced in achieving pure counterflow conditions in THE's.

A further type of fluid flow found in heat exchangers is the crossflow configuration in which the fluid directions are 90° to each other. If the fluid can move about freely while passing through the exchanger, the fluid is said to be mixed.

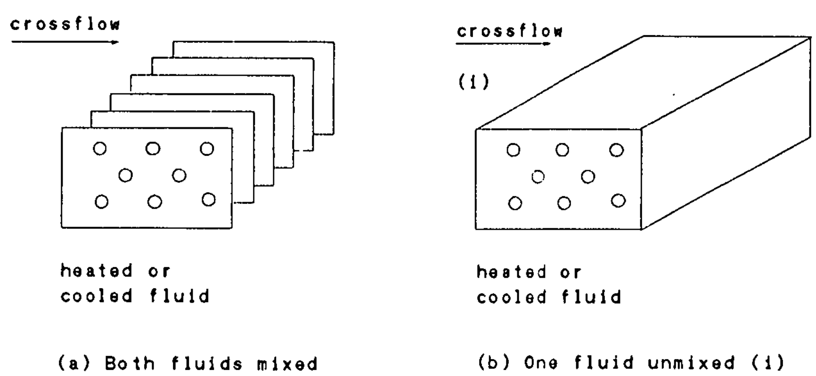


Figure 3.17 Fluid Flow in Crossflow Type Heat Exchangers

Calculation of Q in a crossflow exchanger must include the correction factor F_t after determining type of flow (in other words - mixed or unmixed). Kern (1983:546) has shown by gas flow analysis that the factor of mixed or unmixed fluids has a marked effect on the thermal efficiency of crossflow heat

Examples of crossflow type heat exchangers are charge air coolers and exhaust gas boilers.

Kern (1983:553) also shows by example that the correction factor F_t is greater for unmixed flows than for mixed flows, and for the three true fluid flow patterns: counterflow, crossflow and parallel flow, the decreasing efficiency of parallel and crossflow (mixed and unmixed) applications with a reducing factor F_t.

3.10 Fouling In Heat Exchangers

The performance of heat exchangers designed depends upon the heat transfer surfaces being clean and uncorroded. Should surface deposits be present, thermal resistance increases, resulting in decreased performance. This added resistance must be accounted from when designing heat exchangers and is included as a fouling factor R_d.

The most important factor affecting the fouling characteristics of a fluid, with respect to a certain type of surface, is the fluid velocity and its resultant influence on shearing force and turbulence, laminar layer thickness and residence time close to the heat exchange surface. This means that there must be good velocity and flow distribution over the whole section and if multipass units are used, good velocity between passes.

After calculating the clean overall coefficient from equation (3-5) the dirt factors must be included to calculate the design overall heat transfer coefficient Ud. As can be seen from the design theory for heat exchangers, the fluid velocity and channel (or tube) sizes have a large influence on the Reynolds number and hence the Nusselt number and heat transfer coefficient h_i and h_{io}.

The design overall heat transfer coefficient must be used when sizing a heat exchanger. From the above, it is obvious that a clean heat exchanger will be more effective when first put into service, but performance will gradually reduce as dirt is deposited on the heat transfer surfaces.

The design overall heat transfer coefficient must be used when sizing a heat exchanger. From the above, it is obvious that a clean heat exchanger will be more effective when first put into service, but performance will gradually reduce as dirt is deposited on the heat transfer surfaces.

Again from the Fourier equation $Q = U_d A(LMTD)$ as the heat exchanger becomes fouled, the values of U_d and LMTD will alter. U_c will remain constant if the fluid velocity is not altered by channel restriction. If the LMTD is calculated from the observed temperatures taken when the heat exchanger is first placed into service (clean) and after a given fouling period the R_d (actual) can be calculated from:

$$R_d \quad = \quad 1/U_d - 1/U_c$$

When R_d (deposited) > R_d (allowed), the heat exchanger must be cleaned. The tabulated fouling factors in Table 3.1 are intended to protect the heat exchanger from delivering less than the desired heat load for about a year. The table of fouling factors has been drawn up after evaluation of data from many process conditions and heat transfer applications. (Kern 1983:532)

If a THE is chosen for service with fouling on the tube side it is important to choose as small a diameter tube as possible. Care must be taken as small diameter tubes produce high pressure drops, but there is better flow distribution. It has already been shown in figure 3.10 Single Pass THE Showing Efficiency Losses that stagnations and regions of low velocity occur when the cross baffles are in the shell side of a THE.

PHE's are relatively little affected by fouling due to a good fluid distribution caused by channel chevrons and small plate clearances resulting in good turbulent conditions. Cross (1979:88) that in typical PHE's the hydraulic diameters are small, varying between only 0,0048m and 0,011m.

unproductive pressure drop. Therefore as low a pass system consistent with good heat transfer should be chosen.

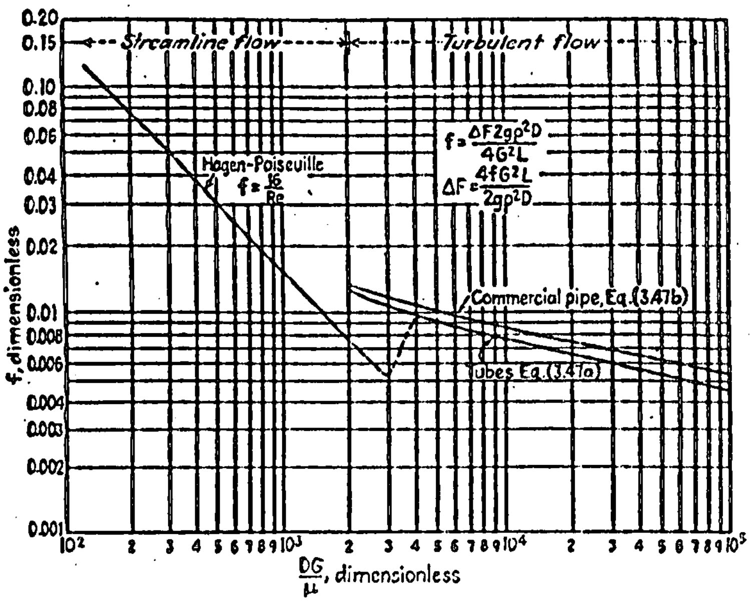


Figure 3.18 Friction Factors for Flow In Tubes and Pipes

(Source: Kern 1983:53)

Kern (1983:51) proves that when fluid flows In a pipe isothermally it undergoes a decrease in pressure. For isothermal turbulent flow this pressure drop is a function of the Reynolds number and in addition to the roughness of the pipe. It has been proved by experiment and referring to figure 3.18 Friction Factors for Flow In Tubes and Pipes - that for streamline flow the Hagen -Poiseuille equation is used to find the friction factor:

$$f = 16/Re \qquad \text{-- (3-29)}$$

Table 3.1 Fouling Factors for Fluids
(Source: Kern 1983: 546)

* Ratings in the last two columns are based on a temperature of the heating medium of 240 to 400°F. If the heating medium temperature is over 400°F, and the cooling medium is known to scale these rating should be modified accordingly.

PETROLEUM FRACTION

Oils (industrial):	
Fuel oil	0.005
Clean recirculating oil	0.001
Machinery and transformer oils	0.001
Quenching oil	0.004
Vegetable oils	0.003
Gases, vapours (industrial):	
Coke-oven gas, manufactured gas	0.01
Diesel engine exhaust gas	0.01
Organic vapours	0.0005
Steam (nonoil) bearing	0.0
Alcohol vapours	0.0
Steam, exhaust (oil bearing from reciprocating engines	0.01
Refrigerating vapours (condensing from reciprocating compressors	0.002
Air	0.002

Liquids (industrial):	
Organic	0.001
Refrigerating liquids, heating, cooling or evaporating	0.001
Brine (cooling)	0.001
Atmospheric distillation units . Residual bottoms, less than 25° API	0.005
Distillate bottoms 25° API or above	0.002
Atmospheric distillation units:	
Overhead untreated vapours	0.0013
Overhead treated vapours	0.003
Side-stream cuts	0.0013
Vacuum distillation units	
Overhead vapours to oil from bubble tower tower (partial condenser)	0.001
From flash pot (no appreciable reflux	0.003

Temperature of heating medium	Up to 240°F		240 - 400°F*	
Temperature of water	125°F or less		Over 125°F	
	Water velocity, fps		Water velocity, fps	
Water	3 ft or less	Over 3 ft	3 ft or less	Over 3 ft
Sea water	0.0005	0.0005	0.001	0.001
Brackish water	0.002	0.001	0.001	0.002
Cooling tower and artificial spray pond				
Treated makeup	0.001	0.001	0.002	0.002
Untreated	0.003	0.003	0.005	0.004
City or well water such as great lakes	0.001	0.001	0.002	0.002
Great lakes	0.001	0.001	0.002	0.002
River water				
Minimum	0.002	0.001	0.003	0.022
Mississippi	0.003	0.002	0.004	0.003
Delaware, Schuylkill	0.003	0.002	0.004	0.003
East River & New York Bay	0.003	0.002	0.004	0.003
Chicago Sanitary Canal	0.008	0.006	0.010	0.008
Muddy or silty	0.003	0.002	0.004	0.003
Hard (over 15 grains/gal)	0.003	0.003	0.005	0.005
Engine jackets	0.001	0.001	0.001	0.001
Distilled	0.0005	0.0005	0.0005	0.0005
Treated boiler feed water	0.001	0.0005	0.001	0.001
Boiler blowdown	0.002	0.002	0.002	0.002

To the right of the transition region for turbulent flow there are two lines, one for commercial pipe and the other for tubes. Tubes have a smoother finish than commercial pipe. There will be no effect on the streamline flow region as the velocity at the tube or pipe wall will be nearly stationary and pressure drop is not influenced by roughness.

Kern (1983:83) quotes the work of Drew, Koo and McAdams where the equation of "f" for fluids in tubes in turbulent flow has been given within 5%:

$$f = 0{,}0014 + 0{,}125/Re^{0{,}32} \qquad \text{-- (3-29a)}$$

Also, for clean commercial iron and steel pipes an equation given by Wilson, McAdams and Seltser within 10% satisfies the friction factor:

$$f = 0{,}0035 + 0{,}264/Re^{0{,}42} \qquad \text{-- (3-29b)}$$

Kern concludes with the Fanning equation which may be modified to give:

$$F = 4fG^2L/2g\rho^2D_e \qquad \text{-- (3-30)}$$

Allowance must be made for the inlet and outlet bends and connections and the allowance of a pressure drop of one velocity head $V^2/2g$ per bend will suffice.

3.11.1 Pressure Drop in Multipass THE's

The pressure drop through the shell of an exchanger is proportional to the number of times the fluid crosses the bundle between baffles and the distance across each bundle each time it is crossed.

The isothermal equation for the pressure drop of a fluid being heated or cooled including entrance and exit losses in a shell is:

$$P_s = fG_s^2D_s(N + 1)/2g\rho D_e o_s \quad \text{kPa}$$

or $$P_s = fG_s^2D_s(N + 1)/5{,}11*10^{10}DeS\phi psf \qquad \text{-- (3-31)}$$

the value of "f" is plotted from the curve in figure 3.18 Friction Factors for Tubes and Pipes.

Kern (1983:148) quotes experiments by Sieder and Tate who have correlated friction factors for fluids being heated or cooled in tubes. They are plotted in dimensional form in figure 3.18 Friction Factors for flow in tubes and pipes and are used in the equation:

$P_t = fG_{t2}Ln/2g\rho D_e o_s$ or $P_t = fG_{t2}Ln/5{,}22*10^{10}D_eS\phi_t$

kPa or psf respectively -- (3-32)

The change in direction introduces an additional pressure drop (called the return loss) P_r and is accounted for by allowing four velocity heads per pass.

Thus $P_r = 4nV^2/2sg$ -- (3-33)

Therefore the total pressure drop on the tube side will be:

$\Delta P_T = \Delta P_t + \Delta P_r$

3.11.2 <u>Pressure Drop in PHE's</u>

The PHE can develop a much higher θ value per pass than any other type of heat exchanger, thus the pressure drop can effectively be used to produce heat transfer. (Heat Exchanger Guide:7)

PHE manufacturer Alfa Laval quote the parameter Je (Jensen Number) as the specific pressure drop and is defined as:

$Je = \Delta P/\theta$(kPa/HTU = 10m wg/htu) -- (3-35)

The channel profile chevrons (high and low) can be chosen to give the desired pressure drop characteristics, the obtuse angled chevrons giving comparatively large pressure drop while the low θ has acute angled chevrons to give smaller pressure drops and less flow resistance.

Pressure drop is approximately proportional to flow/channel. Optimum pressure drop usually lies between Je = 20 - 100 kPa/HTU depending on heat transfer material.

3.12 Summary

3.12.1 The basic theory to heat transfer has changed very little with the THE being first used in the 19th century. The PHE dates back to its first application in 1937.

3.12.2 The pioneer work carried out by Sieder, Tate, Underwood and others is still valid for modern heat exchangers. Development has been to find materials offering better heat transfer characteristics. Extensive experimenting has been carried out to compile tables of heat transfer coefficients, fouling coefficients, friction factors and so on.

3.12.3 It is apparent that flow patterns enhance heat transfer coefficients by improving Reynolds and Nusselt numbers. Prandtl numbers can be obtained from the tables or the dimensionless group $(Ck/\mu)1/3$.

3.12.4 PHE's have many advantages over the THE for certain applications by promoting turbulent flow from plate patterns.

3.12.5 Pressure drop is critical in heat exchanger design as heat transfer is directly dependent on pressure drop. High pressure drop results in good heat transfer with smaller heat exchangers being possible. High pressure requires careful consideration when sizing the circulating pump.

3.12.6 The theory includes counterflow and crossflow type exchangers, with pure counterflow being the most efficient for water/water, oil/water applications.

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3.13 Nomenclature for Terms used in Chapter 3

Symbol	Explanation	Unit
A	Area	m^2
a	area/plate	m^2
C	Mass flow*specific heat	
C_p	Specific heat	kJkg K
D	Diameter	m
D_e	Effective diameter	m
d	Diameter	m
E	Effectiveness	
f	Friction factor	
f_t	Correction factor	
g	Acceleration of gravity	m/sec^2
G	Mass velocity	kg/s m
h_i	Heat transfer coefficient	kW/m^2 K (inner)
h_o	Heat transfer coefficient	kW/m^2 K (outer)
k	Thermal conductivity	W/m K
L	Length	m
LMTD	Log mean temperature diff.	°C
m	Mass flow	kg/s
N	Number of baffles	
n	Number of passes	
Nu	Nusselt Number	
P	Pressure drop	kPa
Pr	Prandtl number	
P_t	Pitch	mm
Q	Heat load	kW
Re	Reynolds number	
R_i	Thermal resistance	w/mK
R_o	Thermal resistance	w/mK
s	Specific gravity	
T_{hi}	Hot fluid inlet temperature	°C
T_{ho}	Hot fluid outlet temperature	°C
t_{ci}	Cold fluid inlet temperature	°C
t_{co}	Cold fluid outlet temperature	°C
$\Delta T1$	(Thi - tco)	°C
$\Delta T2$	(tho - tci)	°C
U_c	Clean coefficient	kW/m^2K
U_d	Design coefficient	kW/m^2K
v	Fluid velocity	m/sec
V	Volume flow	m^3/sec
ξ	Emissivity	
0	Radiation heat coefficient	w/mK
ϕ	Viscosity correction factor	
μ	Dynamic viscosity	kg/ms
ν	Kinematic viscosity	m^2s
ρ	Density	kg/m^3
τ	Shear stress	MPa
θ	HTU	

Subscripts

t - tube, s - shell, e - equivalent, w - wall,
i - inner, o - outer, io - inner referred to outer

CHAPTER FOUR

HEAT RECOVERY UNDER IDEAL CONDITIONS and PLANT DESIGN

4.1 Overview

The chapter covers heat recovery under ideal conditions as laid down in Sulzer bulletin 4-107.059.310 (Appendix C.6.4) The sources of heat recovery on the Roxana Bank are investigated and the design of the plant is based on the temperatures and pressures found in the bulletin.

To ensure good heat recovery, the cooling system is split into a high temperature (HT) and a low temperature (LT) circuit combined with a central cooler.

The specifications are drawn up after investigating the sources of heat losses allowing a boiler feed water system to be designed to enhance the exhaust gas boiler output.

4.2 Heat Recovery Sources Under Ideal Operating Conditions

The possible sources of heat recovery on the Roxana Bank can be divided into the following groups:

4.2.1 Main engine cylinder circulating water which has been designated the high temperature (HT) circuit.

4.2.2 Main engine lubricating oil and charge air cooling water - these exchangers are fitted in series and have been designated the low temperature (LT) circuit.

4.2.3 Main refrigerating plant cooling water condensers.

4.2.4 Main engine exhaust gas after turbocharger (High temperature gas).

4.2.5 Main engine exhaust gas after boiler - Direct contact heat exchanger.

Referring to (Appendix F DMF 001/89), it is important to note that the high temperature and low temperature circuits are common and pass through a central cooler, which maintains the circulating raw water temperature at an outlet temperature of 32°C. The temperature of 32°C has been selected from the Sulzer bulletin 4-107-310 (Appendix C.6.4) recommending an ideal inlet temperature to heat exchangers of 32°C. This temperature of 32°C will ensure that the lubricating oil, charge air and cylinder water will be within the recommended parameters.

4.3 Heat Recovery from High Temperature (HT) Water Circuit

By installing a heat exchanger in the HT circuit to gain full benefit from the high cylinder water outlet temperature, heat recovery in a water/water application is possible on the secondary side by using:

4.3.1 A large quantity of water with a small temperature change,

4.3.2 A small quantity of water with a large temperature change.

The controlling parameters for items 4.3.1 and 4.3.2 are:

(i) pressure drops

(ii) heat exchanger sizes

(iii) pumping costs

Heat recovery from the HT circuit has a low heat value when compared with the high exhaust gas temperature and careful consideration must be given to construction and installation costs correlated against the effects of heat recovered and the influence this has on the improvement of overall plant efficiency.

If a large quantity of water is used t_2 will be further away from T_1 and less heat exchanger surface area will be required due to the larger LMTD. Although this will reduce initial investment, the operating cost will be increased due to the greater quantity of water and the power required to pump this water through the system.

In evaluating the possible heat recovery from the HT circuit, the effect of undercooling the engine must be considered and avoided. Under ideal conditions the heat to be dissipated from the cylinder cooling water is 896 kW, from which the FW distiller requires a heat value of 442 kW for efficient operation. As the temperature program on the feed water side has been obtained from the data in Appendix A.4.1, the following primary and secondary fluid particulars are assumed constant due to steady state main engine operating conditions.

(1) Main engine cylinder cooling water (primary fluid) volume flow of 84 m^3/hour

(2) Main engine cooling water inlet temperature to heat recovery unit of 85°C

(3) Feed water (secondary fluid) temperature program about the heat recovery unit of inlet 43°C and outlet of 73°C.

Using the heat balance equation:

$$Q = MC_p(T_1 - T_2)$$

to find the heat value recovered, a variation of secondary fluid volume flows are substituted for M (mass flow) and plotted against heat recovered. After the FW distiller heat load the amount of recoverable heat is:

896 - 442 = 454 kW

Assuming item (3) remains constant it can be seen from figure 4.1 Volume flow to heat recovered that for a range of 1 m^3 to 11 m^3 a recoverable heat value ranging from 37,8 to 415,8 kW is possible.

All values above 454 kW are discarded due to only 454 kW being available for heat recovery. Therefore the volume flow should not exceed 12 m^3/hour and a volume flow of 10 m^3/hour has been selected as the three way valves are not accurate to within 1°C and this will prevent inadvertently undercooling the engine. The design of 10 m^3/hour volume flow will result in a heat recovery of 378 kW and coupled with the heat recovered from the FW distiller results in a total heat recovery of 136,6 kW per cylinder (490 MJ per hour per cylinder).

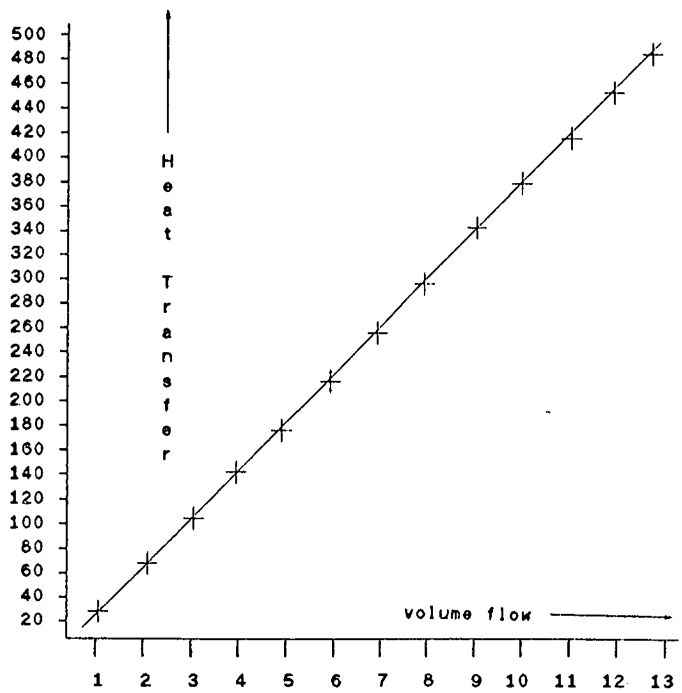


Figure 4.1 Volume Flow to Heat Recovered

This recoverable heat compares favourably with Wartsila's projections (Wartsila Diesel(b) 1983:48) of a similar rated power engine. It should be noted at this stage that Wartsila generally operate with a cylinder outlet temperature of 91°C and an inlet of 85°C. Sulzer Engineer, Mr R M Mueller (Mueller RM (a)) has stated that Sulzer engines may be operated at these temperatures without any adverse effects.

When running at these temperatures and higher, the operating parameters must be kept within close tolerances to prevent thermal overloading of the engines. Gallois (1981:9) has conducted experiments with an SEMT Pielstick PC4 which has been operating quite successfully with a cylinder outlet temperature of 120°C.

Figure 4.3 Heat recovered - main engine cylinder cooling water shows the expected heat recovery from a Wartsila Vasa 32 engine recovering heat from the cylinder water only.

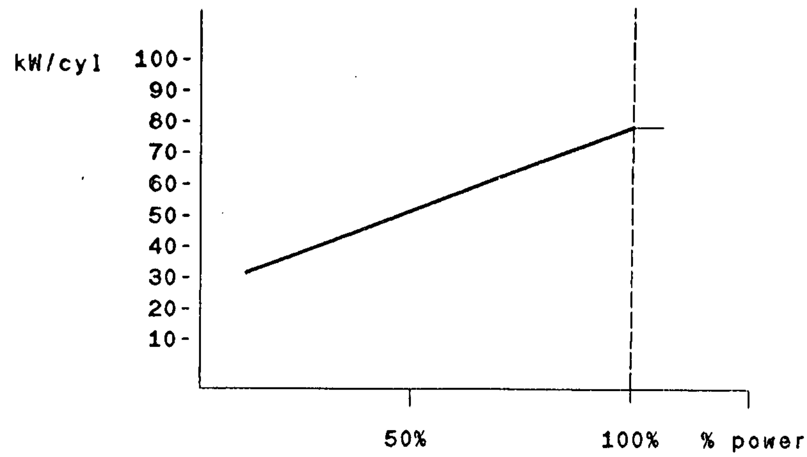


Figure 4.2 Heat Recovered - Main Engine Cylinder Cooling Water

Figure 4.3 Available heat in high temperature charge air shows the effect of increased heat recovery when the high temperature section of the charge air cooler is incorporated into the heat recovery circuit.

Figure 4.3a Heat balance for main propulsion engine shows the effect of increased heat recovery when the high temperature section of the charge air cooler is incorporated into the heat recovery circuit.

Figure 4.3a Heat balance for main propulsion engine shows how the splitting of the cooling water system into HT and LT circuits controls the heat balance of the engine and indicates the recoverable heat from both circuits.

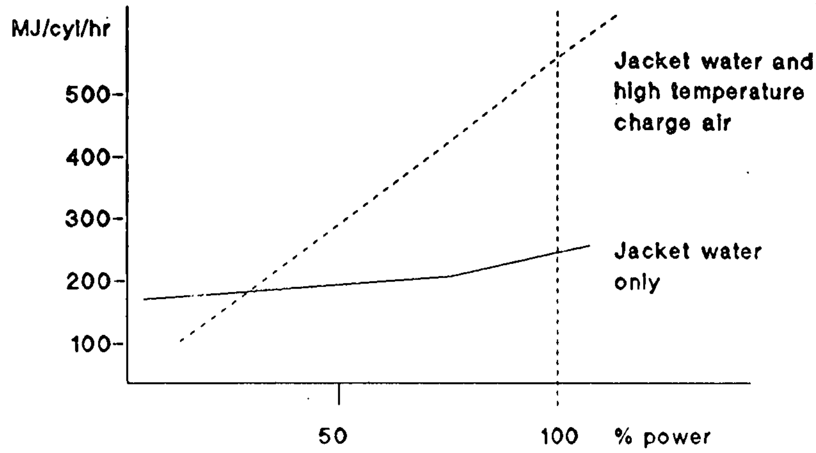


Figure 4.3 Available Heat in High Temperature Charge Air and Jacket Water

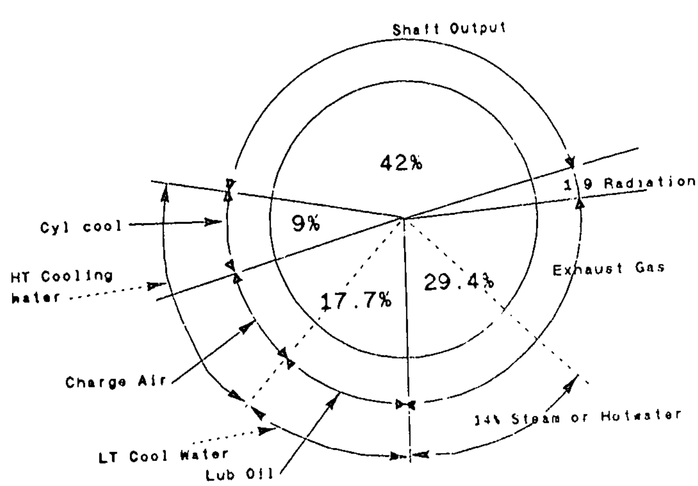


Figure 4.3a Heat Balance for Main Propulsion Engine

The amount of forced draft air into the engine room can be reduced by as much as 80% if the turbocharger air suction is led directly to the outside atmosphere. The compressor suction conditions will then be subject to prevailing external air conditions and an air heater should be utilised to raise the air compressor suction air temperature.

Ideally this heating medium will come from the LT circuit. Such a system will require careful design as the turbocharger suction cannot tolerate large pressure drops in the suction trunk.

A reduction of the engine room forced air supply will lower the electrical load slightly on the main switchboard as the number of running ventilation fans will be reduced.
Considering the equation:

$$Q = MC_p(T_{fi} - T_{fo}) \qquad \text{-- (3-20)}$$

a small mass flow will extract a small amount of heat and in order to achieve effective heat recovery, the feed water flow through the heat recovery unit is in excess of the exhaust gas boiler requirements. Thus with a larger mass flow the heat recovered will be more substantial. When returning the excess feed water to the hot well, it may be used for low temperature heating purposes.

The system design to ensure adequate heat recovery requires the main engine cylinder water outlet to be raised to 85°C. This temperature conforms to Sulzer bulletin 4-107.059.310 (Appendix C.6.4) but could in reality be raised to at least 90°C. Engine inlet temperatures would be in the region of 9° to 10°C lower than the cylinder water outlet temperatures. The temperatures must be maintained within close tolerances to maintain steady state heat recovery.

The LMTD and the pinch point will have a direct influence on the dimensions of a heat recovery unit based on the temperature program in Appendix A.4.1. However should the feed water temperature outlet approach the cylinder water inlet, undercooling of the main engine will result, requiring bypass valves to control the fluid flow in the cylinder water circuit.

Designing a heat recovery system based on a cylinder water outlet temperature of 85°C will require temperature control bypass valves to maintain these raised temperatures.

The engine temperature program is:

Engine cylinder water outlet - 85°C

Engine cylinder water inlet - 75°C

The modification to the cylinder water outlet will involve the elimination of the existing jacket fresh water heat exchanger and the fitting of the Transon THE (Appendix A.4.2a). The Transom THE has been designed considering the temperature program of 85°C outlet and 76°C inlet to the engine and dimensions have been determined by the quantity of heat recovery available. The calculations in Appendix A.4.1 confirm Transheat's dimensions in its place. The high temperature heat recovery unit (HTHRU) will be placed in series with and before the FW distillers.

Full bore flow will be maintained through the HTHRU and distillers, although the existing bypass system must be retained to bypass the distillers for when they are not in use or when maintenance is required on them.

Provision will also be made for the distillers to be placed in series should the HTHRU be placed out of service. The temperature control valves A and B will be retained and will be fully automatic and have the following configuration:

The automatic control valve A will be controlled by fitting a temperature sensor and controller in the engine water discharge pipe and the automatic control valve B will be controlled by fitting the sensor in the engine water inlet pipe. Due regard must be given to low inlet and outlet temperature limits by the fitting of audible and visual alarms.

Initially, the design of the cylinder water circuit included a normally open or normally closed

(NO/NC) valve fitted in the suction pipe before the return branch of the automatic valve B.

The configuration utilising the NO/NC valve will ensure that no cold water from the central cooler will mix with the engine water during start up, resulting in large temperature fluctuations. Engine operating temperatures require a gradual increase to prevent thermal stressing and to ensure gradual expansion of engine components until operating temperatures have been reached.

The temperatures are raised either by a steam heater or a bleed from the auxiliary engine cooling water. When the engine water temperature reaches the operating parameters, the valve will open. Opening of this NO/NC valve will be damper controlled to avoid a sudden surge of cold water mixing with the circulating water.

It is possible that during startup, the central cooler will be fully bypassed, making the use of the sea water circulating pump unnecessary until operating temperatures are reached. At this stage the sea water pump will start either on low speed or with a frequency controller depending on the method employed.

Figure 4.4a Jacket cooling circuit - 1st design shows the cooling water circuit with valves A and B and the NO/NC valve.

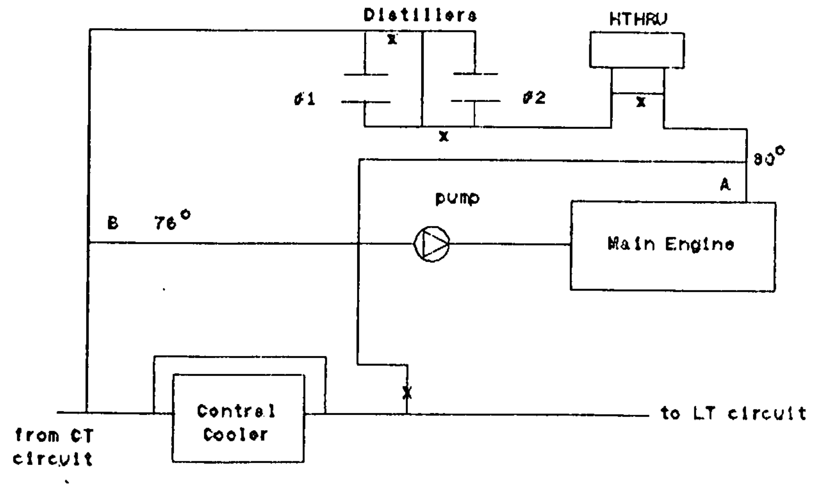


Figure 4.4a Jacket Cooling Circuit - 1st Design

The placing of the automatic valve B as shown in figure 4.4a Jacket cooling circuit - 1st design could result in large temperature fluctuations should the NO/NC valve malfunction, or the central cooler outlet temperature reduces to such an extent that cold water reduces the engine inlet temperatures.

A modification to the circuit requiring the repositioning of the automatic control valve B will give more positive temperature control and eliminates the risk of cold water slugs and undercooling. The NO/NC valve can be dispensed with. There is no risk of pump starvation as the circuit is always kept under a positive suction head from the cylinder water header tank. The position of the automatic valve B is shown in figure 4.4b Modified jacket cooling circuit.

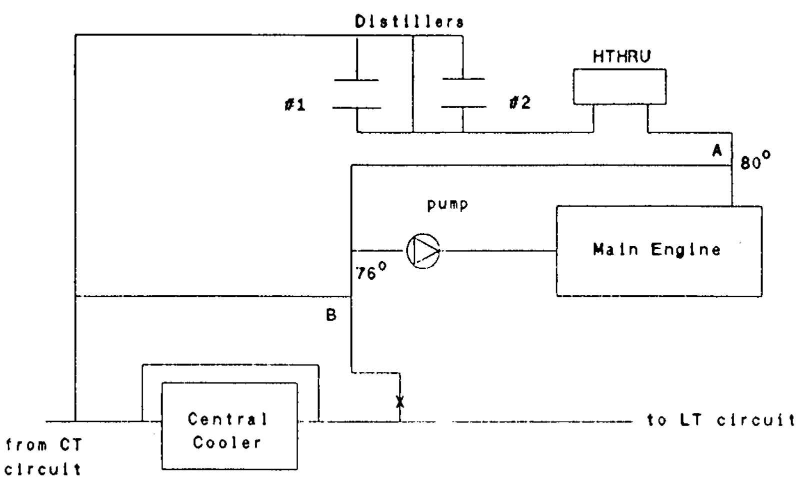


Figure 4.4b Modified Cylinder Cooling Circuit

The existing jacket circulating pump will be retained as the cylinder water flow must be maintained at 84 m^3/hour at all times through the engine. Due to the nature of the design of the total circuit mild steel piping can be used and schedule 40 by 125mm (5") piping will be used as the entire circuit will be circulating fresh water eliminating the corrosion problems normally associated with the use of sea water as a cooling medium.

The design utilising a HT and a LT circuit showing all components including the central cooler proposed for the Roxana Bank and the heat balance with heat recovery and heat dissipation as shown in Appendix F-DMF 001/89; DMF002/89.

The alteration of the jacket water cooling circuit will result in an altered pressure differential as the HTHRU has entirely different characteristics to the now discarded jacket water heat exchanger. As already discussed, the heat exchanger is unsuitable for a volume flow of 10 m^3/hour, and the circuit consists of the main engine, HTHRU, distillers, piping, automatic and isolating valves. Excess heat will be dissipated by the central cooler.

The pressure drops for the proposed system circuit will be:

4.3.1.1 Pressure drop - main engine

4.3.1.2 Pressure drop - #1 FW distiller

4.3.1.3 Pressure drop - HTHRU

4.3.1.4 Valve losses -(9)

4.3.1.5 Pipe length - 20 meters

4.3.1.6 Reducing diameter (1)

4.3.1.7 Increasing diameter (1)

4.3.1.8 Bends (20)

4.3.1.9 Discharge head - 4 meters maximum.

There is a further source of heat recovery based on the FW distiller heat requirements of 442 kW, should the HTHRU be out of service. The two distillers can be operated in series with a total of 884 kW being recovered. The jacket cooling water temperature using this configuration is shown in table 4.1 Heat recovery from cylinder water - distillers from the calculations in Appendix A.4.2.

Table 4.1 Heat Recovery from Cylinder Water - Distillers

#1 FW Distiller			#2 FW Distiller		
Q	T_{in}	T_{out}	Q	T_{in}	T_{out}
442	85	80,4	442	80,4	75,7

It is evident that the #2 FW distiller temperature is close to the ending inlet temperature resulting in complete heat recovery from the cylinder water. However, during periods of low power requirements, temperatures will not reach the proposed program inlet of 85°C as full bore flow must always be maintained through the engine and FW distiller production will fall off and undercooling could result. At this stage the automatic control valves would operate and bypass the heat recovery units and or the FW distillers, to maintain the required engine inlet temperatures.

The use of the two FW distillers in series would require the full heat load from the jacket cooling water and there would be no mixing with the LT water and therefore no HT flow through the central cooler. Operation with a HTHRU and #1 FW distiller would result in a heat surplus and therefore mixing with the LT water will occur as shown in Appendix A.4.3. The design of the primary heat recovery unit is shown in Appendix A.4.1. The initial design was to maintain similar velocities in primary and secondary fluids with the design results as shown.

However, after discussions with Mr Holland of Transheat, it was decided that all heat exchangers would be off the shelf units and of modular design so as to increase or decrease in size if required.

Thus the Transheat unit will be used. It can be seen that from the calculations in Appendix A.4.1 and the computer calculations into Reynolds numbers and Nusselt numbers are similar, small discrepancies coming in because of differing fluid velocities. Pressure drops have also been included in the Transheat calculations and the variations are due to the spacing of the spiral segment plates being placed 40, 60 and 80 mm apart.

4.4 Heat Recovery from the Low Temperature Circuit (LT)

This circuit incorporating the main engine lubricating oil and charge air cooler will have under, under ideal conditions, an outlet temperature a cooling water temperature of 47,2°C. In the Roxana Bank there would be little application for this low grade heat but for vessels operating in arctic conditions, this low grade heat can be used for preheating the combustion air prior to entry into the turbocharger suction.

In this application the suction air should be preheated to at least 35°C and air preheater will be utilised should this system be fitted. Careful consideration must be given to suction trunk pressure drop so that the engine is not starved of air. In this application a plate type heat exchanger is not suitable as the pressure drop on the air side should not fall below 20 kPa (Alfa Laval (a) undated:10). To facilitate a compact unit, a crossflow type heat exchanger will be utilised.

4.5 The Main Refrigerating Plant Condensers

The refrigerating plant operation gives a cooling water temperature rise of 2°C and with a volume flow of 66 m^3/hour per heat exchanger will result in an available heat recovery of approximately 150 kW per condenser.

The refrigerating condenser inlet temperature is based on the circulating water temperature of 32°C. Considering a rise in 2°C the outlet temperature would be 34°C resulting in this heat load of 150 kW.

A further possibility also exists to design the complete raw water system to include the refrigerating condenser cooling water, however the raw water mass flows would become too great - in the order of 170 + [3 * 66] = m^3/hour, resulting in a large pump and central cooler requiring a primary fluid flow of 520 m^3/hour with its associated large pump.

In order to reduce the corrosion possibilities and maintenance periods, the entire refrigerating system should be converted to a raw water system incorporating its own central cooler.

4.6 The Direct Contact Heater

The exhaust gas temperature after the exhaust gas boiler must be in the order of 160° to prevent acid attack on the mild steel exhaust trunk and upper heat exchanger surfaces of the boiler.

Goldstick and Thumann (1983:127) state that a source of heat recovery is found in this low temperature section by the use of a direct contact heater in which the primary water to be heated comes into direct contact with the exhaust gas.

This water extracts latent and sensible heat from the exhaust gas and in turn transfers heat to the secondary fluid through a counterflow heat exchanger. However, Goldstick and Thumann (1983:134) have stated that to obtain full benefit from this low grade heat, the entering primary fluid to the sprayers must be of a sufficiently low temperature. Figures quoted are below the expected feed water temperature of the feed tank outlet and therefore the full effect of the direct contact between the exhaust gas and the primary circulating water would not be viable due to the expected feed water temperature of 40°C.

A further negative aspect of the direct contact heater is that due to the low exhaust gas temperatures, corrosion resistant materials must be used in the exhaust gas boiler, exhaust gas trunk and the entire direct contact heater, resulting in high initial cost.

4.7 Excess Feed water Heat Recovery

The boiler should ideally produce approximately 800 kPa (absolute) at a rate of 0,6944 kg/s. This is the projected steam consumption at present, including the fish meal plant. At 10 m^3/hour volume flow, the excess feed water flow is in the order of 2 kg/s returning to the feed tank. To gain full effect of the heat recovery unit, this mass flow of 2 kg/s at 73°C must be reduced in temperature before entry into the feed tank to attain a suitable heat balance about the feed tank.

The feed water excess flow mass flow and considerable high temperature can be utilised to preheat the fuel oil (blended fuel) and lubricating oil purifier circulating oil. A further use for this feed water as a heating medium is preheating the sea water feed to the second FW distiller which will reduce the heating steam consumption.

Referring to Appendix F(DMF006/89), the position of the heat exchangers is shown in relation to the proposed system along with the expected temperature and fluid flow programs.In order to remove the heat from the excess feed water and gain maximum benefit, the heat exchangers will be fitted in the following order:

4.7.1 Lubricating oil purifier main engine oil preheater

4.7.2 Blended fuel preheater for settling tank

4.7.3 #2 FW distiller sea water feed water heater.

4.7.1 <u>Lubricating Oil Preheater (#1)</u>

The main engine lubricating oil is put through a centrifugal purifier to maintain good quality oil by extracting water and separable solids. The throughput is based on the formula: 20% flow through separator in relation to nominal capacity where:

$$V(\text{nominal}) - 1{,}2 \text{ to } 1{,}5\ (P)\ \text{kW} \qquad \text{-- (4-2)}$$

where P is the total engine output. V(nominal) is in litres/hour.

(Source: Wartsila Diesel(b)1989:40).

Thus throughput = (20*1,5*3180)/100

= 954 lites/hour

which corresponds with the data supplied by Mr R M Mueller of Sulzer Brothers. (Mueller R M (b) 21 July 1989).

With the centrifuging temperature at 80°C (recommend 75° to 85°C), preheating can be carried out by two methods:

from approximately 50°C in the first stage

and 65° to 80°C in the second stage

In preheating the oil, steam or electrical consumption will be reduced, but will necessitate the installation of two heaters with increased pressure drop and installation costs.

Initially the design of the preheater was based on a double pipe heat exchanger and calculations show that the flow for lubricating oil was laminar with a low Reynolds number. Using an iterative process as shown in Appendix A.4.5, the heat exchanger length was too large demonstrating the inherent inefficiency of double pipe heat exchangers and the importance of maintaining fluid flows in the turbulent region.

Using the heat exchanger NTU method gives a calculated effectiveness of 0,54 and a heat exchanger surface area of 5,069m^2 which again results in an unacceptable length.

From the data provided by Appendix A4.4 it can be seen that the PHE heat exchange surface area is 2,28 m^2 - this indicating the influence of both sides of the plate being effective. The temperature program is:

Feed water : 73° to 72°C - flow rate 2,000 kg/s

Lubricating oil : 50° to 65°C - flow rate 0,264 kg/s

with a heat load of 7,83 kW.

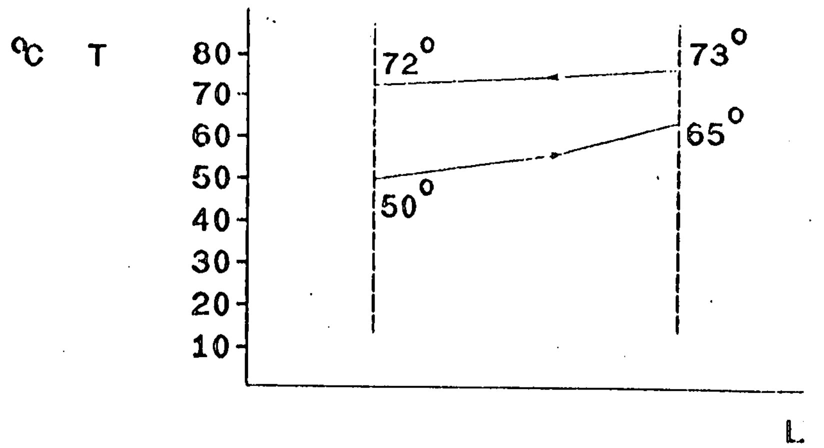


Figure 4.5 Temperature Program Lubricating Oil Preheater

By nature of its design the PHE can create turbulent flow with Reynolds numbers as low as 10 due to small hydraulic diameters and plate ribbed pattern. Cross (1979:12) has determined through experiments that turbulent flow will also reduce the incidence of fouling.

The second stage of preheating will require a rise in temperature of 15°C and ideally the heating medium should be steam as steam has a high latent heat value. With a further 8 kW required to maintain 80°C at the separator, the total steam required can be calculated: (assuming a dryness fraction of 0,85)

Heat required for 550 kPa $= h_f + (xh_{fg})$

$= 656 + (0{,}85*2096)$

$= \underline{2437{,}6\ \text{kJ/kg}}$

therefore, the mass of steam required for the lubricating oil preheater #2 is,

steam mass flow $= 10{,}58/2437{,}6$ (kJ/s//kJ/kg)

$= \underline{0{,}0043\ \text{kg/s}}$

4.7.2 Lubricating Oil Preheater #2

From the Appendix A.4.5 the overall transfer coefficient ranges from 113 to 340 W/m^2K (Heater Transfer:242). These values have been correlated from past experience with the evaluation of a wide range of heat exchangers using different fluid/fluid applications.

Steam from the range will be reduced to 300 kPa absolute (200 kPa gauge) with an inlet temperature of 133,5°C into the heater. With a total heat value of $(h_f + xh_{fg})$ after 550 - - 300 kPa reducing valve:

Q(550 kPa abs)	=	656 + (0,85 * 2096)	-- (4-3)
	=	<u>2437,6 kJ/kg</u>	
With a heat load of Q	=	$MC_p(T_1 - T_2)$	-- (3-20)
	=	0,264 * 2,003 * (85 - 65)	
	=	<u>10 58 kW</u>	

<u>Confirming the mass of steam required as 0,0043 kg/s (9,62 kg/hr)</u>

Steam as a heating medium can introduce several problems. Hot steam condensate is fairly corrosive and should be routed through the tubes in order to protect the steel shell. However, to increase the heat transfer area, and to reduce the length of the heat exchanger to acceptable dimensions the number of tubes will be 37 and using the design of heat exchangers for the heat recovery unit, the outer shell dimension will be based on an internal diameter of 102 mm.

Routing the lubricating oil through the shell will result in a low velocity as shown:

Flow area (shell)	=	$(\pi/4 * 0{,}102^2) - 37\,(\pi/4 * 0{,}007^2)$	
	=	$\underline{0{,}00675\ m^2}$	-- (4-4)

From $Q = V * A$ $\quad V = 1{,}1/(3600 * 0{,}00675)$

$= \underline{0{,}0453\ m/s}$

If the lubricating oil flows through the tubes the velocity will be from:

$$Q = V * A \qquad \text{-- (4-5)}$$

$$1{,}1/3600 = V*\pi/4*0{,}007^2*37$$

$$= \underline{0{,}2146\ m/s}$$

With a velocity of 0,2146 m/s the flow must be verified to ascertain whether the flow is turbulent or laminar. Referring to Appendix A.4.5 with a Reynolds number of 21,69 confirming laminar flow.

To prevent condensation from accumulating in the heat exchanger and to maintain uniform flow the condensate line must be fitted with discretion. If the pressure inside the heat exchanger drops below that of atmosphere, the condensate will fill up the shell shutting off the heating surface area, resulting in the steam not condensing and retaining its inlet pressure to blow out the condensate which will cause a cyclic operation.

The heat transfer coefficients associated with the condensation of steam are very high when compared with any studied thus far (Kern 1982:164). All heating surfaces employ a relatively air free stream, thus a value of 8500 W/m^2K will be used for the condensation of steam (Kern 1983:164). Thus $h_i = h_o = h_io = 8500\ W/m^2K$.

Since steam is an isothermally condensing fluid, the true temperature difference Δt and LMTD will be identical.

To reduce pressure drop of the steam the flow of condensate in the shell must be unrestricted and therefore no baffles will be used - the fluid will be mixed in the shell.

The pressure drop including entrance and end losses through an exchanger can be calculated by taking the pressure drop for steam as calculated from the inlet steam rate and flow area of the shell in the usual manner by the equation

$$Ps = (fG_s^2D_s[n+1])/2g\rho d_{e\phi}s \qquad \text{-- (3-31)}$$

Pressure drop in the tubes (lubricating oil) is found from the equation:

$$\Delta P_t = \Delta P_{tu} + \Delta P_r \qquad \text{-- (3-34)}$$

When there is a supply of exhaust steam and process steam an analysis of the relevant temperature programs will determine the heating supply. In this particular application the only supply of exhaust steam in the form of condensate is from the fish meal plant returning at a rate of approximately 0,33 kg/s. Although exhaust steam possesses a high latent heat the exhaust steam is of limited process value as the saturation temperature is in the region of 100°C. The lower temperature as opposed to the saturation temperature of 133,5°C (at 300 kPa abs) will result in larger heat transfer surface areas. There is thus a limited use for exhaust steam. Furthermore, should the fish meal plant be out of use for whatever reason, the supply of exhaust steam will be terminated resulting in no heating medium to the lubricating oil and the blended fuel preheaters.

The possibility of using the exhaust steam to raise the boiler feed water from 73 to 95°C (Δ_t = 22°C will be evaluated, with an initial liquid saturated heat value increase of 392 - 305 = 84 kJ/kg).

4.7.3 Blended Fuel Preheater (#1)

From the heat load balance equation, the feed water outlet temperature from the lubricating oil preheater is calculated at 72°C, and therefore heat is available for preheating the fuel oil.

The Roxana Bank fuel preparation system consists of a settling tank of $24m^3$ capacity and a service tank of $10m^3$ capacity. The existing conditions make no provision for heating of the settling tank contents.

Due to the nature of blended fuel, circulation should be continuous to prevent any possibility of fuel separation due to differing densities and viscosities. The fuel passes from the settling tank through a separator into the service tank. At present the purifier heater is electrically powered.

The blended fuel consumption averages at 7200 litres/24 hours resulting in this amount being used from the settling tank. Assuming filling of the tank only occurs once in 24 hours, from the basic heat balance the temperature of the fuel after filling will be 50°C. To reduce the load on the separator heater the fuel will be preheated to 65°C. Using data found when determining the lubricating oil preheater size and information supplied by Alsa Engineering (Appendix A.4.6) for a PHE, the heat exchanger surface area is $2{,}04m^2$ and the temperature program is:

Feed water:	72,1° --- 71,1°C flow rate 2,000 kg/s		
Blend fuel:	50,0° --- 65,8°C flow rate 0,264 kg/s		
	Heat load:	8,24 kW	

The following values must be used when calculating fuel heating equipment:

(1) Fuel viscosity 180 cSt at 50°C - temperature rise in heater 65°C (Add 10% + 5kW to compensate for radiation losses).

(2) Recommended viscosity at fuel pumps is 14 cSt.

(3) The required minimum capacity of the inline fuel heater is: (Project Guide:46)

$$p(kW) = \{m(l/h) * t(^\circ C)\}/1700 \qquad \text{-- (4-6)}$$

where m is the specific fuel consumption, resulting in a minimum capacity of 15,53 kW. This heater will remain either electrical or steam.

(4) The separator preheater is placed in the circuit before the fuel oil separator and the required minimum capacity is found from the empirical formula in (3) and using a temperature rise of 15°C the capacity is 5,83 kW.

(5) The fuel oil separator outlet is calculated from the empirical formula:

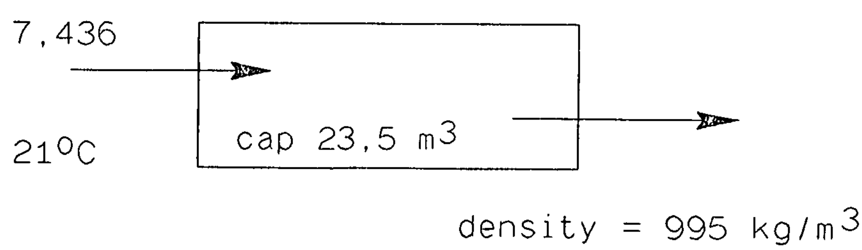
$$V(\text{nom } l/h) = (24P)/A \text{ kW} \qquad \text{-- (4-7)}$$

(where A = 30% for fuel viscosity of 180 cSt 50°C)

Thus the throughput is 2544 l/hour (0,707 l/second). From the heat balance and total heat required of 659948 kJ the heat required for a 24 hour operation would be 7,63 kW resulting in a mass flow rate of 0,264 kg/s.

The settling tank particulars are: capacity 23,5 m^3 blended fuel.

Engine fuel consumption = 7500 litres in 16 hours. It is likely that the engine will run on gas oil for the remainder of the 24 hours as no fishing is carried out at night and if the engine is operated at below 40% it should run on gas oil. This will be verified during actual operating conditions in November 1989.



At start of refilling = 16,037 at 65°C

Oil from bunker tank = 7,436 at 21°C

Resultant temperature after filling :

$$(23{,}5 * T_r) = (16{,}037 * 65) + (7{,}346 * 21)$$

$$T_r = 51°C$$

The heat loss through the shell of the ship must be taken into consideration when calculating heat load required by the blended fuel heater. The heat exchanger must therefore be sized accordingly. Heating is required to raise the temperature to 60°C. Total heat required by fuel

$$Q = 23{,}5*995*2{,}016*(65 - 51)$$

$$= \underline{659948 \text{ kJ}}$$

If the vessel only consumes blended fuel for 16 hours, steam is not required after the change over. Heating steam should be supplied for 16 hours however a heating time of 24 hours has been used in the calculation.

This oil will be circulated and the separator will draw from the settling tank. From the above it is apparent that the load on the separator heater will vary from a maximum when the fuel settling tank is at 50°C to a minimum when the temperature is 65°C. In order to calculate the interim heat load, the following equation may be used:

$$dQ/dt = k(\theta_1 - \theta_2) \qquad \text{-- (4-7)}$$

to calculate the temperature at any time which in turn will enable the heat load

$Q = mC_p\Delta_t$ to be calculated.

As the heat load varies the heat required to preheat the blended fuel will vary. For the fuel oil separator to supply fuel as it is consumed the throughput should be in the region of 0,1 kg/s which varies substantially from the empirical formula for V(nominal). Should a flow 0,699 kg/s pass through the separator there will be large overflow return to the settling tank. It has been proved from operating experience that fuel purification is superior when the throughput is reduced and with a mass flow of 0,1 kg/s the heat load will be 3kW.

At commencement of the heating cycle the fuel will be raised from 50 to 85°C resulting in a heat load of $Q = mC_p\Delta_t$, being Q = 7,03 kW. Using the steam pressure of 550 kPa (absolute), the mass of steam at commencement of heating will be:

$$\text{mass steam (kg/s)} = \frac{7{,}03}{2995{,}2}$$

$$= \underline{0{,}0023 \text{ kg/s}}$$

The design of the blended fuel preheater will be the same as the lubricating oil preheater. Specifications are shown in Appendix A.4.8.

4.7.4 #2 FW Distiller Sea Water Heater

With a feed water temperature of 71,08°C and a mass flow of 2,2 kg/s there is still available heat. The #2 FW distiller is operated using steam as the feed water heating medium with the heat exchanger process taking place within the distiller.

A proposed alternative to reduce steam consumption and hence reduce fuel consumption is to preheat the sea water to 63°C before entering the distiller shell at which it would flash off as the absolute pressure in the shell is -0,95kPa. Alsa Engineering have supplied the following data for a PHE and is shown in Appendix A.4.12, 12a, 12b.

For a heat load of 349,7 kW

Table 4.2 Sea Water Preheater Design Data

TEMPERATURE				MASS FLOW					PD		
SW In	SW out	FW In	FW out	SW	FW	LMTD	NTU	Area	SW	FW	k
20	63,4	71,2	30	2,0	2,0	8,2	5,4	7,92	58	56	5,39

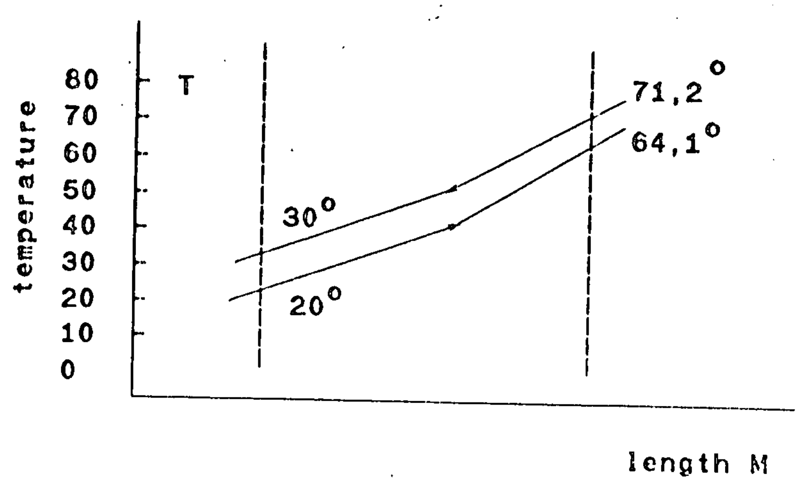


Figure 4.6 Temperature Program - Sea Water Heater

From the design data it can be seen that the heat exchanger surface area is large, i.e. 7,92m. This result is from a large thermal length or theta (θ) value of 5,38 NTU. Furthermore the cost of this unit will be increased by at least 2,5 X as titanium is thus more expensive than stainless steel. However as the Titanium is corrosion resistant the expected life of the PHE is very extensive.

The sea water can be raised to 63°C as the increased turbulence created by the ribbed pattern of the heat exchanger plates will reduce the incidence of fouling and scaling resulting in a more efficient heat exchanger with minimal maintenance. As there will be a large capital outlay, this must be carefully correlated against fuel saved by not using heating steam to operate the #2 FW distiller. As the production of water is 12,4 tonnes/24 hours and the flow to the distiller is 172 tons/24 hours, a large amount of this feed water will be dumped to the sea.

This feed water will also be used as the ejector water to maintain the operating shell vacuum. Notice must be made of the fact that if heating steam is used a large amount of heat is also lost as the excess feed water is carried away.

The fresh water feed to the feed tank must be reduced in temperature to ensure that the feed tank suction temperature to the feed water heat recovery unit is low enough to ensure adequate heat recovery.

4.7.5 Design of #2 Feed Water Heater (73° - 93°C)

Before the mass flow rate of the heater (feed water) is determined, the boiler capacity must be calculated along with the plant steam requirements. The initial heat to produce steam is from the exhaust gas from the main engine. This heat is a low grade heat and assuming steam generation from the exhaust heat only, the total heat available allowing an exhaust gas outlet temperature from the boiler tubes to be 160°C to prevent corrosion, is calculated from the equation:

$$Q = mC_p\Delta_t \qquad \text{--(3-20)}$$

$$= 5{,}83 * 1{,}016 * (350 - 160)$$

$$= \underline{1125{,}4 \text{ kW}} \quad (\text{kJ/s})$$

The specific heat capacity calculation of the flue gas is found in Appendix A.4.8.

If the boiler pressure is to operate at an outlet pressure of 550 kPa absolute and assuming a dryness fraction of 0,85 the steam output will be :

$$m = 1125{,}4/(694 + [0{,}85 * 2075)$$

$$= \underline{0{,}4578 \text{ kg/s}}$$

The maximum rate that the fish meal plant requires is 1300 kg/h (0,36 kg/s), which means that the boiler output will supply the total needs of the vessel at an efficiency of 100%.

The effectiveness of the boiler must be calculated later so that the actual output on exhaust gas heating can be determined.

The temperature range of the feed water and condensate will give a LMTD of 4°C (Appendix A.4.9). When calculating for a double pipe heat exchanger unit the overall heat transfer coefficient is low resulting in a large heat transfer area. To overcome the difficulties that could be experienced with hot condensate corrosion a PHE unit has been evaluated using stainless steel plates and with an average heat transfer coefficient of 4,50 kW/m^{2}K, the heat exchanger dimensions are within acceptable limits, bearing in mind the limited space available.

(It should be borne in mind at this stage that with correct boiler water and condensate treatment, the incidence of metal corrosion will be greatly reduced in feed water piping and heat exchangers).

Referring to equation (3-27) θ is used to define the thermal length of a heat transfer operation. A high θ value involving a large number of HTU's, represents a duty where a fluid is cooled or heated by a temperature change which is large in relation to the

$$\text{LMTD} - \quad \theta_1 = (t_1 - t_o)/\text{LMTD} \qquad \text{-- (3-27)}$$

From the calculations for the #2 Feed Water Heater, the θ value is 5 HTU. The initial calculation for the PHE Type A3 size resulting in 18 plates has not taken any pressure drops or flow rates into account.

When using a pressure drop the specific area is 5m^2 resulting in a plate increase to 31 plates. Calculations and specifications for the PHE model PO give a plate stack of 56 plates (corrected for pressure drops).

Assessing the calculations and specifications of the Type A3 and Type PO as follows:

Type A3 and Type PO as follows:

Table 4.2a Principal Dimensions PHE Type A3, PO

height	width	length	conn	area	plate	flow	
mm	mm	mm	mm	m	m	m^3/h	
690	185	232,5	25	5,2	0,058	9	Type A3
460	180	320	42	2,98	0,032	14	Type PO

To keep the fluid velocities within acceptable limits as calculated in the standard $Q = V*A$ formula the 25 mm connection Type A3 should be selected.

From the heat exchanger calculation for the feed water heater (Appendix A.4.9), the excessive length is due to the LMTD being low and low primary and secondary fluid velocity flows resulting in low Reynolds numbers in the turbulent zone but insufficiently high to promote good fluid thermal conductivity values, even though flow rates and velocities are almost matching, normally ideal conditions for heat transfer.

The use of a THE will improve the fluid velocity in the tubes, but the velocity in the outer shell will be negligible resulting in laminar flow and poor heat transfer. Passing the condensate through the shell will reduce the velocity even further.

Considering the design of the PHE's gives a large thermal length but a smaller surface area. It should be noted here that the value of 4 500 W/m^2K has been taken as an average of 1500 to 6 000 W/m^2K specified in the PHE specifications (Alfa Laval(a) undated:38) also allowing for 15% fouling resistance. Correcting for pressure drop will increase the number of plates in the heat exchanger as shown in Appendix A.4.12. The specifications of the main heat exchanger has now been completed for the heat recovery section and assuming ideal main engine operating conditions on the Roxana Bank heat recovery will be possible with heat exchangers as specified in Table 4.3 Heat exchanger specifications. The existing lubricating oil and charge air heat exchangers have been retained as they are efficient enough for requirements and low temperature heat recovery is not envisaged at this stage.

Table 4.3a Heat Exchanger Specifications

	terms	units	Exch 1	Exch 2	Exch 3	Exch 4
Primary Fluid	volflow	m^3/h	84	7,4	7,4	7,4
	T_i	°C	85	73	72	71,2
	T_o	°C	81,1	72	71,1	29,3
	bulk	°C	83,1	72,5	71,6	50,3
	t	°C	3,92	1,0	1,1	41,9
	ρ	kg/m^3	972,8	975,8	976,2	989,9
	c_p	kJ/kgK	4,1966	4,1790	4,1785	4,1777
	massflo	kg/s	22,68	2,0	2,0	2,0
	p	kPa	130,6	7,6	39	56,6
	fluid		cylwater	fd water	fd water	fd water
Secondary fluid	volflow	m^3/h	10	1,08	1,07	7,0
	T_i	°C	40	50	50	20
	T_o	°C	73	65,8	65,8	64,1
	bulk	°C	56,5	57,9	57,9	42,1
	t	°C	33	15,8	15,8	44,1
	ρ	kg/m^3	987,1	876,8	884,6	1025,9
	C_p	kJ/kgK	4,1775	1,978	1,978	3,967
	massflo	kg/s	2,74	0,264	0,264	2,00
	p	kPa	120,3	2,75	14,4	7,01
	fluid		fd water	lub oil	fuel oil	sea water

The combined fluid specifications for the actual heat exchanger design are found in the figure 4.3b Combined specifications.

Table 4.3b Combined Specifications

	terms	unit	Exch 1	Exch 2	Exch 3	Exch 4
COMBINED	heat	kW	373	8,25	8,25	349,5
	θ	NTU	1,398	1,23	1,24	5,37
	LMTD	°C	23,6	12,8	11,8	8,2
	A/L	m/m	6,57	2,28	2,04	7,92
	flow	type	counter	counter	counter	counter
	plates	No	-	21	19	68
	tubes	No	6 x 2	-	-	-
	passes		one	one	one	one
	applic	fluid	wat/wat	wat/oil	wat/fuel	wat/sea

4.7.6 Heat Recovery with Low Feed Water Flow

An alternative to splitting the cooling water circuits into an HT and LT system, is the use of the existing boiler feed pump and circulating water through an exchanger with a bleed off from the jacket cylinder water and then through the high temperature section of the charge air cooler.

With the charge air entering the intercooler at approximately 125°C, and considering a pinch point of 10°C, the feed water could be raised to 115°C should the high temperature section have sufficient length and heat transfer surface area.

The jacket heat recovery heat exchanger will be fed with approximately a quarter of the volume flow through the jacket system to reduce heat exchanger size and keep pressure drop to minimum.

The HT section of the charge air cooler will be modified by designing end covers to split the feed water and sea water systems. The physical dimensions of the end covers and water guides will be designed when temperature and heat load problems have been completed.

Should calculations show that this system will work on the Roxana Bank the conversion can be done relatively quickly and at low cost, bearing in mind that no modifications will be carried out to the sea water cooling system.

The design in Appendix A.4.16 shows that while a suitable arrangement can be found in the jacket system with a temperature rise up to 80°C, but with the existing tube diameters, the fluid velocity only reaches 0,42 m/sec and the Reynolds number is bordering on the transition region (2300 - 20000), resulting in a low heat transfer coefficient and a large thermal length. This system would also require a high and low temperature circuit with central cooling.

The system is however, not feasible due to the low fluid velocities.

4.8 The Exhaust Gas Boiler

The use of the exhaust gas boiler in diesel engine installations has become widespread to generate steam for electrical power and general shipboard use - for example fuel heating.

Recovering heat from exhaust gas is possible by lowering the exhaust gas temperature to approximately 170°C by heat transfer. The limitation of 170°C is imposed to prevent acid attack on the mild steel tubes and trunks of the exhaust system.

Large quantities of steam are generally only possible with large exhaust gas mass flows as the heat value is of low quality and hence large mass flows will increase the heat available when the equation:

$$Q = MC_p(T_i - T_o)$$

is used. It can be seen that if the specific heat value and the Δ_t remain constant, Q will change as the mass flow changes.

Modern installations use highly efficient boilers extracting maximum heat available by fitting economisers and superheaters.

It is necessary for the purpose of this study to investigate the exhaust gas temperature differential and exhaust gas mass flows to ascertain the effectiveness of the boiler.

4.9 The Conversion to the HT and LT System

The cooling water system will be split into a high temperature and a low temperature circuit which will have a common suction and discharge section. The temperature will be controlled by the central coolers. (Appendix F DMF001/89) The circulating water will be fresh water which will be cooled by sea water in the central cooler. The central cooler will be manufactured from titanium which will extend the life of the cooling system almost indefinitely.

In order to ensure that the heat exchangers are correctly utilised and efficiently operated, the circulating water inlet temperature will be raised to 32°C. The lubricating oil and charge air cooler will be retained with the jacket cooler being discarded and a heat recovery unit being installed in series with the FW distillers. The full circuit is shown in Appendix F DMF003/89).

The pumping alternatives for the HT and LT circuits are:

4.9.1.1 One 170 m^3/hour circulating pump with standby duplicate supplying the system as shown in Appendix F - DMF004/89.

4.9.1.2 One 84 m^3/hour pump with standby duplicate supplying the HT system with one 83 m^3/hour pump with standby duplicate supplying the LT circuit as shown in Appendix F - DMF005/89.

The advantages of a single pump system would be a reduction in the number of pumps required and less piping. Piping would be sized to maintain the correct flow rates and fluid quantities throughout the system.

The advantages of a single pump system would be a reduction in the number of pumps required and less piping. Piping would be sized to maintain the correct flow rates and fluid quantities throughout the system.

Fitting two circulating pumps (HT and LT) is more practical as there will be more control over the pumped fluid and the temperature control will be more positive. It is anticipated that the power consumption will be similar for both applications.

Wartsila (Wartsila Diesel (b) 1989:55) and Sulzer have advocated dedicated pump operation - an example can be found on the Unicorn vessel MV BORDER - for HT and LT systems. The jacket cooling water pumps will be retained for the HT system and two pumps will be supplied for the LT system.

The existing sea water pumps will be retained for the sea water cooling of the central cooler. In this application for the Roxana Bank, the central cooler has been designed for maximum heat dissipation and maximum fluid mass flows.

However, provision is made for low load operation and supply of the FW distillers.

Central cooling systems are becoming more popular in use in engine rooms of a variety of output powers for the following reasons:

4.9.2.1 Splitting the system into HT and LT circuits allows more control over heat recovery and leads to greater heat recovery, particularly when larger output powers are evaluated.

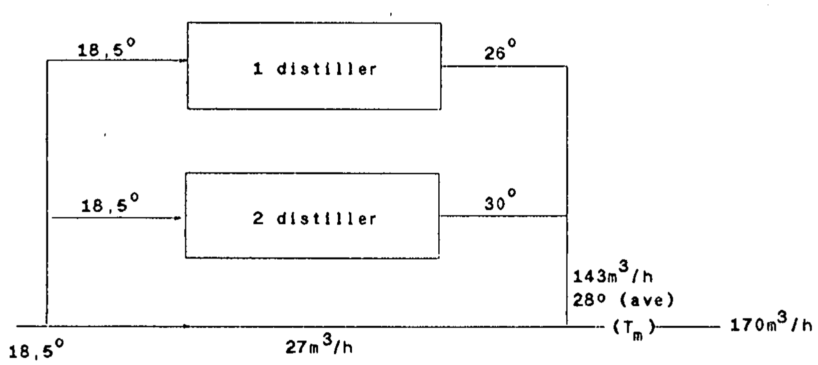
4.9.2.2 The system in its entirety is fresh water cooled resulting in minimal fouling of heat transfer areas which in turn reduces maintenance and cleaning costs.

4.9.2.3 The sea water circuit of the central cooler is kept very short with the pump and cooler as close to the hull as possible.

4.9.2.4 The use of fresh water means that cheaper materials can be used in the heat recovery units and heat exchangers including piping and valves and will hence reduce capital costs.

The design of the central cooling system for the Roxana Bank requires that the circulating fresh water for the LT circuit be maintained at 32°C if heat recovery circuit is required. This, however, presents problems when incorporating the cooling water for the FW distillers as the distillers require a volume flow of approximately 1,3 x heating water flow (75 m^3/h) per distiller which in turn will require a pump of at least 150^3/hour. The distillers could be incorporated in the fresh water system if option 4.2.1.1 is used. Since option 4.2.1.2 has been selected, the water cooling for the distillers will be kept as sea water. There is adequate pump capacity for this service and the distillers will be piped in series with the central cooler, the distillers being in parallel with each other and splitting the flow of 170 m^3/hour.

From the temperature program of the distillers taken during actual operating conditions (Roxana Bank 09/11/89), the temperature of the sea water after mixing is:



$$(170 * T_m) = (143 * 28) + (27 * 18{,}5)$$

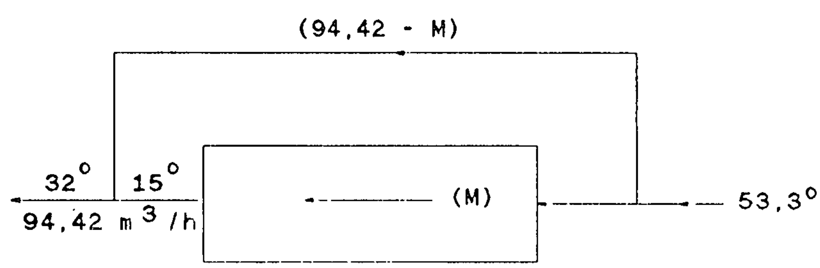
$$T_m = 26{,}5°C$$

Thus the sea water temperature into the central cooler (secondary fluid) is 26,5°C. The design of the central cooler requires to be dimensioned for a fluid inlet temperature (secondary) of 32°C. Again, the central cooler will also operate with inlet temperatures as flow as 15°C with the FW distillers shut down.

To prevent undercooling of the hot fluid (primary fluid) a bypass valve will be fitted to the hot fluid circuit with set point 32°C to maintain the LT inlet temperature. Bypassing the central cooler will tend to underutilise the cooler and low mass flows will result in an inefficient heat exchanger.

With a hot fluid inlet temperature of 47,2°C and a low cold fluid inlet temperature of 15°C due to the thermodynamic laws, the cold fluid maximum outlet temperatures will be 47,2°C and the hot fluid minimum temperature will be 15°C assuming heat exchangers of infinite length. At this temperature it is obvious that there will be minimal fluid passing through the central cooler. Referring to Appendix A.4.11 central cooler temperature and volume flow programs, the maximum inlet temperature (primary side) is 53,3°C.

If the outlet is to be 32°C then the maximum flow through the central cooler will be : (Assuming sea water temperature of 15°C)



It is assumed that due to the heat exchanger being designed for maximum heat dissipation (in other words with 94,42 m^3/h flow) the thermal length is such that the fluid passing through the heat exchanger will reduce to 15°C.

Verifying this value to calculate the dissipated heat:

$$Q = MC_p\Delta_{trw} \qquad \text{-- (3-20)}$$

$$Q = (52{,}5 * 990 * 4{,}18 * (53{,}3 - 15)/3600$$

$$= \underline{2311{,}8 \text{ kW}}$$

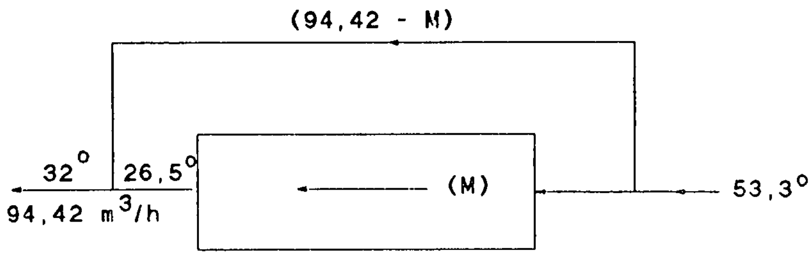
which compares favourably with the values in Appendix A.4.11. Calculating of the sea water outlet temperature at a rated 126 m^3/hour will give T_o:

$$Q = MC_p\Delta tsw$$

$$2311{,}8 = \{126 * 1031 * 3{,}95 * (T_o - 15)\}/3600$$

$$T_o = \underline{31{,}22^\circ C}$$

For an inlet temperature of 26,5°C to the central cooler will result in a more efficient utilisation of the heat exchanger as shown:



$$(94{,}42 * 32) = \{(94{,}42 - M) * 53{,}3\} + (M * 26{,}5)$$

$$M = \underline{75\ m^3/hour}$$

Again heat dissipated by the central cooler will be:

$$Q = MC_p\Delta_{trw} \qquad \text{-- (3-20)}$$

$$Q = \{75 * 990 * 4{,}18 * (53{,}3 - 26{,}5)\}/3600$$

$$Q = \underline{3210{,}4\ kW}$$

Verification of the sea water temperature outlet will be:

$$Q = MC_p\Delta_{tsw}$$

$$2310{,}4 = \{126 * 1031 * 3{,}95 * (T_o - 26{,}5)\}/3600$$

$$T_o = \underline{42{,}7^\circ C}$$

It should be noted at this stage that the volume flow (secondary fluid) is 126 m^3/hour. From the design requirements of a PHE, the primary fluid or fluid recovering heat should be in the order of 1,3 to 1,4 x the hot fluid volume flow, resulting in 126 m^3/hour.

Therefore this flow will be used in the FW distillers. Splitting the flow will show that the volume flow of 63 m^3/hour per distiller does not satisfy the design requirements of the distillers of minimum 1,3 x the primary fluid flow (81,5 m^3/h), and is in fact 1,15 times the volume flow.

The flow of 126 m^3/hour will be retained as the design is on the central cooler and a flow of 170 m/hour will result in a more inefficient heat exchanger, including a greater pressure drop. As the FW distillers are being retained, it is anticipated that with the reduced sea water flow will raise the outlet temperature of this sea water by approximately 1° to 1,5 °C.

Table 4.4 Central cooler specifications shows the various combinations of primary and secondary fluid flows and temperature programs and the influence these have on heat exchanger (thermal length). The initial design conditions 1 and 2 are based on the temperature program using an inlet temperature of 30 °C.

For the calculation of thermal length:

$$\theta_1 = (t_i - t_o)/\Delta_{lm} \qquad \text{-- (3-19)}$$

$$= (53{,}3 - 32)/4 = \underline{5{,}33}$$

Comparing the 0_1 value with the temperature program in condition 3, the thermal length 0_2:

$$\theta_2 = (t_i - t_o)/\Delta_{lm}$$

$$= (53{,}3 - 32)/12{,}3$$

$$= \underline{1{,}73}$$

Table 4.4 Central Cooler Specification

	terms	units	CONDITIONS 1	2	3	4	5	6	7
HOT FLUID	T_{in}	°C	53,3	47,5	53,3	47,5	53,3	53,3	47,2
	T_{out}	°C	32	16,2	32	27,5	26,4	15	26,4
	masflo	kg/s	26,2	22,6	26,2	22,6	26,2	26,2	26,2
	PHE(M)	kg/s	26,2	11,1	17,8	17,5	20,6	14,4	19,1
	P	kPa	132	101	121	92,3	130	108	98
COLD FLUID	T_{in}	°C	30	16	22	16	26,4	15	26,4
	T_{out}	°C	46,3	46,3	38,3	35,4	42,7	31,2	37,9
	masflo	kg/s	36,1	24,7	36,1	24,7	36,1	36,1	36,1
	P	kPa	199	98,1	197	96,8	199	199	199
COMB	LMTD	°C	3,99	0,56	12,3	11,8	6,3	27,1	9,68
	Area	m^2	100	100	27,8	27,8	27,8	27,8	27,8
	Load	kW	2330	1467	2330	1464	2311	2310	1661
	Plates		138	138	39	39	39	39	39

θ_2 gives a value of 1,73 which significantly affects the heat transfer area. Using the basic formula:

$$\theta = (kA)/MC_p \qquad \text{-- (3-21)}$$

will indicate the large difference in area between conditions 1,2 and 3. Referring to conditions 1 and 2, the outlet temperature primary fluid is very close to the inlet temperature secondary fluid giving a very small pinch point value and a large thermal length which in turn gives a large heat transfer area. To reduce this heat exchanger in size and also significantly reduce the capital cost, the heat exchanger must be designed on a sea temperature of 22 °C maximum.

The Roxana Bank will then be restricted to an area of operation south of Cape Town and Port Elizabeth which is the designated fishing area preferred by Irvin and Johnson trawlers. Should the vessel ever be required to move to warmer fishing grounds, then the PHE can be expanded simply by adding pairs of plates to suit the new temperature program. Adding and/or removing plates is unique to PHE's and indicates their versatility.

The effect of the dramatic increase in the heat exchange area and hence capital cost are shown in a steam turbine/exhaust gas heating design where a low pinch point results in good heat recovery and hence power output, but requires a larger surface area and thus increased price. Increasing the pinch point reduces the surface area quite considerably with a small percentage loss in power. (Anon 1978b:69) Referring to Appendix C.6.2.a, heat recovered to surface area illustrates this factor where a reduction of heat transfer surface area from 4,0 to 3,0 m^2 (reduction of 25%) reduces the heat recovered from 205 to 185 kW (reduction of 9,8%).

It is imperative that the cost involved in the conversion remains at a minimum.

The central cooler will account for a large portion of the conversion costs and since the unit material is titanium, this cost is high.

The design of the central cooler is therefore based on a sea temperature of 22°C and the Roxana Bank must operate accordingly. The effects of an 8° rise in the sea temperature (for calculation purposes) is shown in Table 4.4 Heat exchanger specifications.

The decision to restrict the central cooler secondary fluid inlet temperature to 22°C requires the evaluation of two items of plant:

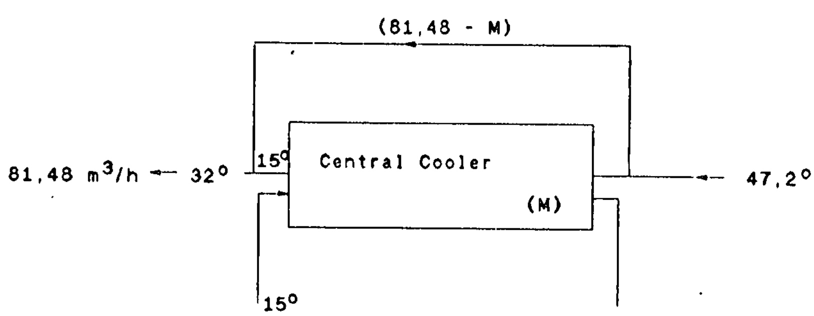
4.9.3.1 The alteration to the central cooler circulating pumps

4.9.3.2 The rerouting and supply source of the cooling water for the FW distillers.

4.10 Central Cooler Circulating Pumps

The maximum heat to be dissipated will be 2330 kW with an attendant volume flow of 94,42 m/hour, and the maximum sea water inlet temperature of 22°C. The minimum heat load to be dissipated will be 1477 kW with the attendant volume flow of 81,48 m^3/hour. It is important that the sea water pump be sized for the 1,35 x the maximum volume flow, in other words 126m^3/hour. Based on a factor of 1,35 the sea water pump volume flow for a fresh water volume flow of 81,48 m^3/hour would be 110 m^3/hour. However with the sea temperature reducing the 15°C and referring to the pump curve in Appendix A.4.13, a volume flow of 80 m^3/hour will be acceptable resulting in the following temperature program taking into account reduced flow and secondary fluid inlet temperature:

secondary fluid inlet = 15°c



$(M_t * H_{tm})$ = $\{(81{,}48 - M) * h_{fi}\} + (M * h_{fo})$ -- (4-1)

$(81{,}48 * 32)$ = $\{(81{,}48 - M) * 47{,}2\} + (M * 15)$

M = <u>38,46 m^3/hour</u>

Therefore the heat to be dissipated at this temperature program:

$Q = MC_p\Delta_{trw}$ -- (3-20)

$Q = \{38{,}46 * 997{,}2 * 4{,}1786 * (47{,}2 - 150)\}/3600$

Q = <u>1433,3 kW</u>

Based on this heat load, the rise in temperature of the sea water will be:

$$Q = MC_p\Delta_{tsw}$$

$$1433,4 = \{80 * 1032 * 3,998 * (T_o - 15)\}/3600$$

$$T_o = \underline{30,6°C}$$

The reduced flow will result in a pressure drop which is in the acceptable limits for the pump application at reduced speed and output. A titanium heat exchanger type PHE model A15 - 8FM having 39 plates will be used. The dedicated pump will have an output of 125 m^3/hour.

Referring to Appendix A.4.12a, it can be seen that the Overall heat transfer coefficient is 5,747 kW/m^2K, (service conditions) reflecting the high efficiency of a PHE.

4.11 Main Engine Cooling System - Central Cooler Design

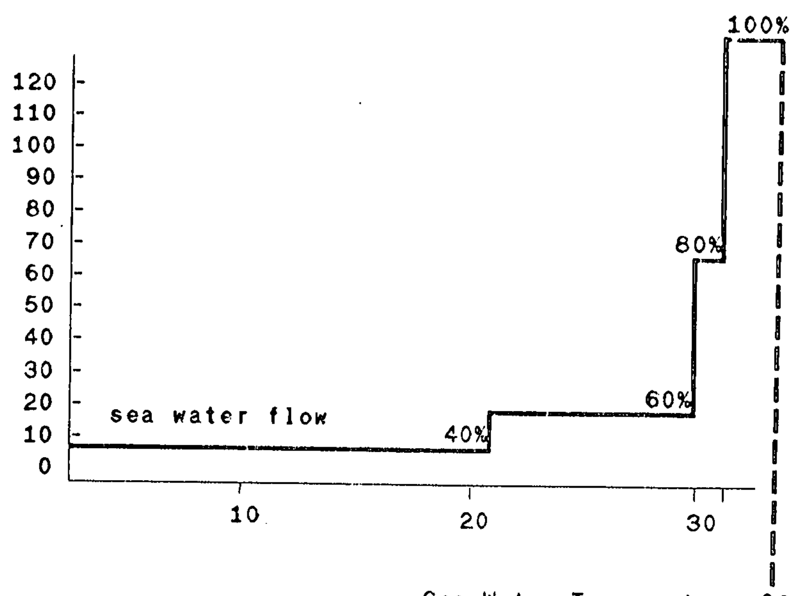
When designing the main engine cooling system careful consideration must be given to the HT and LT volume flows for all possible conditions. The heat exchangers must be designed for an ideal outlet temperature of 32°C. (Appendix C.6.4) In the proposed system for the main engine cooling the HT and LT fluid is raw water and this fluid is temperature controlled by using a three way control valve bypassing the central cooler and the primary fluid is sea water at an average temperature of 20°C. This is the average temperature of the sea water for the area of operation. However the heat exchanger must be designed for a primary inlet water temperature of 32°C and a primary cooling fluid volume flow should be in the order of 1,3 to 1,4 times the secondary fluid volume flow.

In this application the raw water is designated the secondary fluid and the sea water the primary fluid.

From the data calculated in Appendix A.4.11, the maximum volume flow through the central cooler is 94,42 m^3/h with no heat recovery taking place. The design data for the central cooler is found in Appendix A.4.12a and b.

The volume flow required in the central cooler is 81,48 m³/h with proposed full heat recovery, and in order to prevent undercooling of the secondary fluid due to the effects of the primary fluid volume flow and lower sea water temperature, the primary fluid circulating pump will be a two speed pump, using a reduced volume flow for a lower sea water temperature with some automatic switch in the system to bring in the higher speed.

The central cooler primary fluid flow system must be designed to accommodate sea temperatures ranging from 15° -- 32°C by varying the volume flow. By adapting sea water flow to suit sea temperature, considerable savings can be made in power consumption. From figures based on a hypothetical, medium sized ocean going vessel, a graph showing how slight changes in the sea water temperature can effect flow requirements can be drawn.



(Source: Alfa Laval (b) undated)

Figure 4.7 Drops in Sea Water Reauirements with Reducing Temperature

Ideally, as sea water temperature alone does not affect primary fluid flow requirements, changes

In engine load will also affect central cooler heat load, good combinations could be:

(i) Two single speed pumps in parallel, resulting in two flow combinations.

(ii) Two pumps in parallel, one single speed and one two speed resulting in three flow combinations.

(iii) Two pumps in parallel, both two speed, resulting in four flow combinations.

As the vessel operates in waters where the sea water temperatures range from 15° to 23°C a three combination installation will be used with automatic switching.

An optimum cooling system is achieved when the annual pumping energy cost and the investment cost are lowest, resulting in the lowest total cost. Alfa Laval have developed a system which evaluates cooling requirements, flow requirements, operating conditions producing a detailed specification called Cooling Water System Optimization (CWSO).

4.11.1 Central Cooler Primary Fluid System

To ensure a cost effective system with minimum power consumption and maximum effective cooling of the engine raw water an optimum system for primary fluid flow must be designed.

In order to eliminate corrosion and erosion problems as well as fouling, a titanium plate PHE will be used. It is important to note that a titanium PHE is about double the price of a standard stainless steel unit of the same size.

Assuming a pressure drop of 75 Kpa, the maximum head (100%) will be 300 kPa with a maximum volume flow (100%) of 130 m^3/h. With a 4-pole motor and a 50 hz supply the pump revolutions will be :

$$N = 60f/P \qquad - (18)$$

where f is the frequency

P is the number of pairs of poles

$$N = 60 * 50/2$$

$= \quad 1500$ rev/min

and when taking into account a 1% slip (phenomena of induction motors) the pump speed will be 1450 rev/min.

Two pumps in parallel service give design capacity of 100% at a pressure head corresponding to the pressure drop in the system.

The pump combinations are now analysed based on a graph (see Appendix A.4.13) plotting the maximum required capacity (100%) to the maximum required head (100%). Thus the design point PD is located at 100%. The system curve is peculiar to the installed primary system and will take into account pipe and valve losses, heat exchanger pressure drops and static heads including pipe friction losses. As the equipment is installed below the waterline there is no static suction lift and thus Po - 0.

Referring to calculations in Appendix A.4.14 and making the following assumptions:

4.11.1.1 induction motor slip 1%

4.11.1.2 pump efficiency is 70%

4.11.1.3 pump law $\{Q/ND^3\}$ as valid

4.11.1.4 pump law $\{\rho gH/N^2D^2\}$ as valid

4.11.1.5 power (shaft) as $P = \rho gQH/\xi$

4.11.1.6 minimum heat to be dissipated as 1490 kW

the optimum use of the cooling water pumps can be evaluated.

Analysing Appendix A.4.14 single pump operation 370 rev/min is not feasible due to the low pressure head value. Single pump operation at 990 rev/min gives a relatively low pressure head value and a corresponding low power consumption. From the calculations taking into account effects of head loss due to valves and the heat exchanger pressure drop (see Appendix A.4.13) a pump speed of 990 rev/min will be suitable.

An alternative to the two speed pump with its complicated two and four pole switching and extra mass resulting in more power consumption, is using two smaller pumps in parallel with each other resulting in an increased volume flow, but maintaining a steady head. A similar switching device will be used.

Although the central cooler must be designed for an operating temperature of 32°C (secondary fluid inlet temperature), the usual sea water temperature in the area of operation is normally in the region of 210°C which would necessitate a lower mass flow rate in order to dissipate the same heat load.

A variation to the energy saving devices apart from selectable pump speeds is the application of frequency converters. Centrifugal loads all have square law speed torque characteristics which means that the torque increases linearly with speed: In other words $N \propto T^2$.

For a system with no static lift, the following characteristics apply:

$$Q \propto N$$

$$N \propto T^2$$

$$P \propto N^3$$

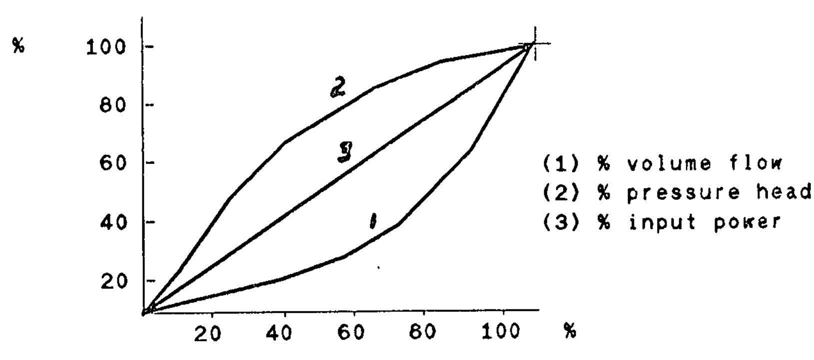


Figure 4.8 Affinity Laws for Centrifugal Loads

Due to the varying sea temperatures heat dissipation loads and hence secondary fluid flows it is impossible to size the centrifugal pumps for a particular condition and equipment must be designed to handle peak demands of a system. However, with full heat recovery taking place, the central cooler will be running at excess capacity.

It is important that at all times, full bore flow conditions are maintained as restrictions in the discharge pipe increase power requirements (in other words throttling in on a discharge valve to reduce flow), which also increases pump pressure and decreases the pump efficiency.

Throttling of the pump discharge valve also alters the system curve. Figure 4.9 Typical 75 kW applications shows the effect of throttling in the discharge valve on the system curve for a typical 75 kW application. The system curve is determined by two components - the static lift and the friction head, which corresponds to the centrifugal law that pressure varies with the square of the flow or speed.

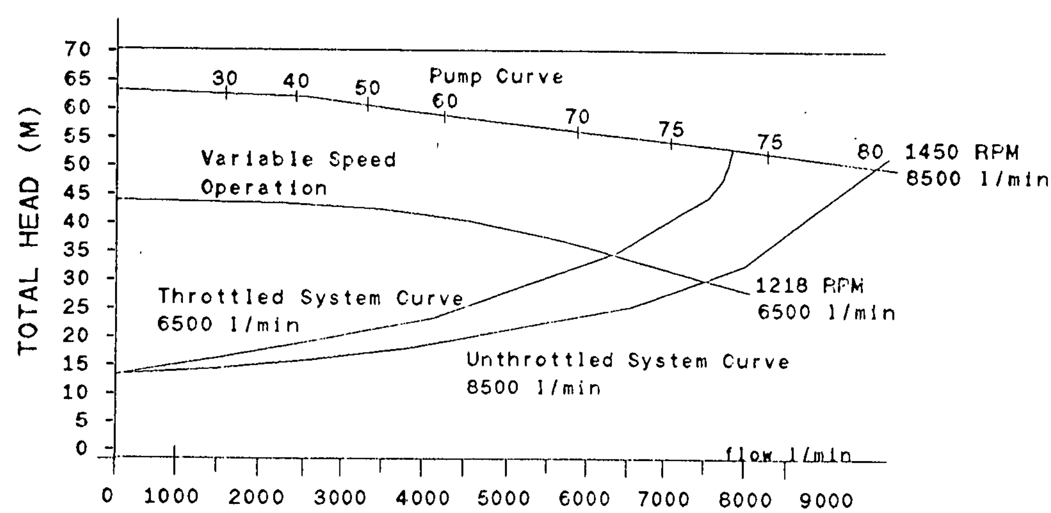


Figure 4.9 Typical 75 kW Application for Centrifugal Pump

Gibson (1989:11) has calculated that by applying a variable speed (frequency) control the wire to water savings show a dramatic improvement over a valve throttling operation.

Apart from making the pumping equipment extremely flexible, the frequency inverters can be used during starting, where the starting torque, which is normally very high can be reduced with the attendant reduced starting loads on alternators and electrical load. Frequency inverters have the ability to handle various signals such as air or electric signals.

With this system, the pump speed should not be reduced to the extent that the sea water becomes the minor fluid and hence the controlling fluid and cannot control the circulating raw water to within the required parameters, which would allow the main engine to overheat. As shown with the selectable pump operation and pump speed reduction in volume flow reduces the power consumption.

Application of the formula:

$$N = 60f/P$$

will demonstrate the effectiveness of frequency control.

PHE design for the central cooler is planned to accommodate maximum flow and maximum temperature conditions. However, the vessel operates in conditions of 22°C and with the heat recovery units in operation, minimum flow conditions are apparent. The design is thus for maximum flow and after recalculating for a secondary fluid flow inlet temperature of 23°C inlet conditions. The owners of the Roxana Bank have the option of storing the extra plates or designating the trawler to operate within the defined fishing zones to suit the smaller heat exchanger.

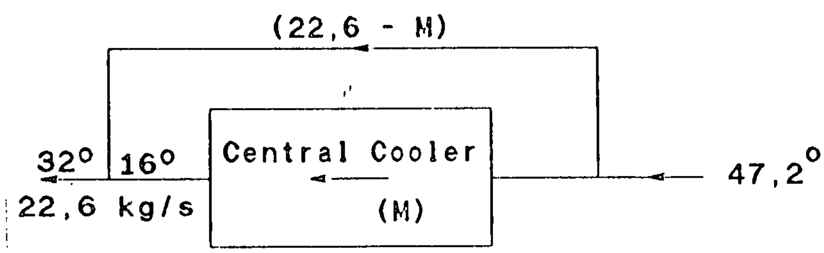
From Appendix A.4.12 it will be apparent that from the reduction in heating/cooling surface area the secondary fluid temperature rise will be reduced as will the temperature drop of the primary fluid.

The temperature program requires a bypass valve to be fitted to the central cooler in order to maintain an outlet temperature of 32°C. Mass flow through the heat exchanger must not be reduced to such an extent that the flow becomes laminar resulting in excessive fouling.

The temperature controlled bypass valve (three way valve) will be fitted to the primary fluid system (engine raw water).

During maximum flow conditions, the temperature program will ensure that the primary fluid system outlet temperature will be 30°C resulting in minimal bypass conditions. When the secondary fluid inlet temperature drops to 16 °C the outlet temperature of the primary fluid is well below 30°C resulting in a large heat load and severe undercooling of the primary fluid. As the primary fluid heat load is 1490 kW, the fluid passing through the PHE must be adjusted to suit the lower heat load.

Assuming a secondary fluid inlet temperature of 16°C, the heat exchanger laws dictate that the primary fluid outlet will be not lower than 16°C. Basing the heat balance about the central cooler on this premise, the following flow rates will apply.



The heat balance calculation for the Central Cooler:

Ref	32°C	h_f	=	134,21 kJ/kg
	47,2°C	hf	=	197,89 kJ/kg
	16°C	h_f	=	67,63 kJ/kg

(Source: Appendix D.1.1)

$$(22{,}6 * 134{,}21) = \{(22{,}6 - M) * 197{,}89\} + (M * 67{,}63)$$

from which M = <u>11,05 kg/s</u>

The result shows 11,05 kg/s flowing through the central cooler and 11,55 kg/s bypassing it. The bypass method is superior to throttling in on the heat exchanger inlet valve as throttling would increase power consumption (see Figure 4.9 Typical 75 kW application) and would increase fluid velocities. Throttling would present a hit and miss temperature control as opposed to the automatic control of the thermostatic valve.

4.12 Revised FW Distiller Sea Water Circuit

With the restriction placed on the sea water temperature inlet to the central cooler renders the original system combining the cooling of the distillers with the central cooler obsolete, and requiring a separate system for the FW distillers.

To incorporate the FW distillers into the fresh water system is not feasible as the outlet temperature from the distiller coolers is in the region of 28° to 30°C, and the secondary fluid volume flow (maximum) through the central cooler does not satisfy the volume flow requirements of the FW distillers. An alternative to the two pump system designed for the HT and LT circuits is to use the single pump of 170 m^3/hour and place the distillers in series with the central cooler fresh water circuit.

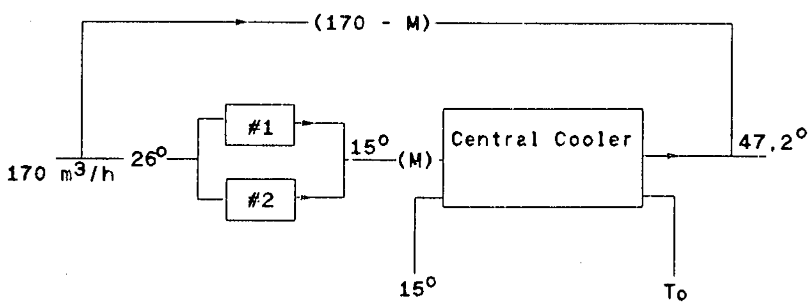


Figure 4.10 Alternative FW Distiller Cooling (Fresh Water)

The arrangement shown in figure 4.10 Alternative FW distiller cooling (fresh water) will allow (from calculation) a volume flow of 122m^3/hour (61 m^3/hour each) to the FW distillers. This quantity is still short of the required volume flow for the distillers and the design lends itself to temperature fluctuations should alterations occur in the operating conditions for the distillers.

To maintain an adequate sea water supply to the distillers, there will be a completely separate sea water pumping system utilising the existing 170 m^3/hour sea water pump.

Provision will be made to use this pump as an emergency pump for the central cooler in event of central cooler pump failure. The effect of this will allow a single pump installation for the central cooler.

4.13 Summary

4.13.1 The identification of heat losses are found in the main engine cooling systems and exhaust gas outlet pipe. To ensure efficient heat recovery, the main engine cooling systems are split into a high temperature and a low temperature system.

4.13.2 The main engine jacket outlet temperatures are to be raised to 85°C and with an expected differential of 10°C the inlet to the engine will be approximately 76°C.

4.13.3 To ensure that undercooling of the engine does not occur, three temperature control valves are required to control fluid temperatures to control set points. The system of heat exchangers designed will reduce the electrical load on the electrical generators.

4.13.4 The central cooler secondary fluid requirements will be varied due to the sea water inlet temperature variations and heat load fluctuations which will be dependent on heat recovery requirements.

4.13.5 The heat recovery prospects are found in the high temperature heat recovery unit and in the excess feed water flow returning to the hotwell. The hotwell outlet temperature is critical to the efficient operation of the heat recovery system. Efficient heat recovery is possible with steady state operating conditions.

4.13.6 The specifications for the heat exchangers have been designed for the maximum heat recovery.

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CHAPTER FIVE

Roxana Bank - Vessel Operating Conditions

5.1 Overview

The Roxana Bank is a 2500 Gross Register Tonne stern trawler built in Gdynia, Poland in 1977, fitted with a fish processing factory and freezer holds to store the frozen fish. The duration of the voyages average 45 days. The machinery plant has been designed so that the main engine which is directly coupled to the gearbox supplies sufficient power for propulsion and electrical generation.

The auxiliary heat exchangers associated with the main engine are all sea water cooled, the inlet temperature being dependent on the prevailing sea temperature.

The main engine runs continuously for all operations at a steady 500 rev/min.

Primary heat recovery on the Roxana Bank at this stage involves the FW distiller which is connected in series and before the cylinder water heat exchanger.

The fuel used is a blend supplied by Mobil Oil of South Africa (Cape Town) which has specifications as found in Appendix B.5.1 which requires heating to maintain a viscosity of 4-6 cSt at the fuel pumps. Fuel heating is by electricity which consumes 112 kW.

Steam supply is from a composite boiler being heated by a combination of blended fuel and main engine exhaust gas. Steam is used in the fish meal plant and FW distillers.

5.2 Main Propulsion Unit

The main propulsion engine is a SULZER 6ZL 40/48 four stroke single acting diesel engine developing 3198 kW at 530 rev/min. Power is transmitted through a reduction gearbox to the output shaft at 160 rev/min driving a controllable pitch propellor. The specific fuel consumption from trial data is 156 gr/hp/hr + 5% when running on diesel oil. The engine is non reversible and runs continuously when steaming and fishing, the power requirements adjusted by setting pitch angles on the propellor.

The engine is turbocharged, the compressor supplying air to the main engine through an air cooler. The exhaust gas is led from the turbocharger to an exhaust gas boiler to generate steam from where it passes to atmosphere.

The engine operating temperatures are kept within close parameters by heat exchangers and temperature control valves, the heat exchangers dissipating the generated heat.

The actual operating parameters are shown in Table 5.2 Actual engine operating temperatures. The secondary cooling medium is sea water which supplies the lubricating oil cooler, cylinder cooling water, charge air cooler (intercooler) and small heat exchangers such as the gearbox oil cooler, controllable pitch propeller oil cooler. The two latter heat exchangers will not form part of the heat balance equations or heat recovery investigation as their contribution to the heat recovery program will be minimal and therefore not cost effective. (Oil mass flow rates low and temperature differentials small).

The present cooling system is set up as shown in figure 5.1 Sea water circulating system - Roxana Bank.

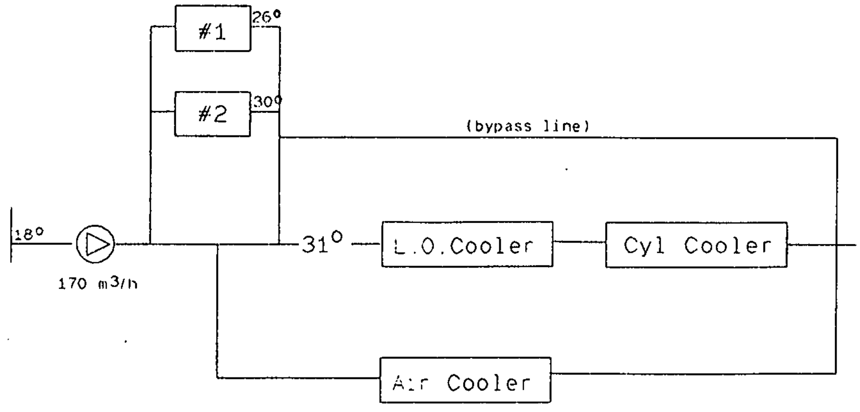


Figure 5.1 Sea Water Circulating System - Roxana Bank

Table 5.1 Technical Data for Auxiliary Equipment

Heat Exchanger			Lub. Oil	Cylinder	Charge Air
Volume Flow		m^3/h	72	84	at 85%
primary	in	°C	66,4	85	power
	out	°C	55	75,8	
temperature	bulk	°C	60,7	80,4	
secondary	in	°C	32	36,1	32
	out	°C	36,1	44,7	42,3
temperature	bulk	°C	34,1	40,4	37,2
density		kg/m^3	864	973,8	1,177
mass flow		kg/s	17,28	22,72	5,83
specific heat		kJ/kgK	2,046	4,193	1,005
heat dissipated		kW	411	875	990
pump head		kPa	600	250	air evaluated at atm press
ΔP engine		kPa			
LMTD		°C	26,4	40	

The sea water pump supplies the heat exchangers as follows:

lubricating oil, cylinder cooler	-	87 m^3/hour
charge air cooler	-	83 m^3/hour
giving a total of	-	170m^3/hour

at a head of 180 kPa.

Table 5.2 Engine Operating Temperatures - Roxana Bank

Heat Exchanger			Lub. Oil	Cylinder	Charge Air
Volume Flow		m^3/h	72	84	at 85%
primary	in	°C	66,4	85	power
	out	°C	55	75,8	
temperature	bulk	°C	60,7	80,4	
secondary	in	°C	32	36,1	32
	out	°C	36,1	44,7	42,3
temperature	bulk	°C	34,1	40,4	37,2
density		kg/m^3	864	973,8	1,177
mass flow		kg/s	17,28	22,72	5,83
specific heat		kJ/kgK	2,046	4,193	1,005
heat dissipated		kW	411	875	990
pump head		kPa	600	250	air evaluated at atm press
ΔP engine		kPa			
LMTD		°C	26,4	40	

Condition 1: Vessel steaming - fish meal plant, factory off

Condition 2: Vessel trawling - fish meal plant, factory operating

Condition 3: Vessel trawling - fish meal plant, factory off

Condition 4: Vessel steaming - fish meal plant, factory operating

Condition 5: Vessel drifting - fish meal plant, factory off

5.3 Jacket Cooling System

The fluid conditions pertaining to the cylinder water heat exchanger are:

Primary fluid - sea water inlet temperature 18°C, tube side, 2 pass.

Secondary fluid - fresh water inlet temperature 68°C shell side, single pass with baffles.

The fresh water circuit includes the #1 and #2 FW distillers, automatic valves and pumps as shown in figure 5.2a Cylinder cooling water circuit - Roxana Bank.

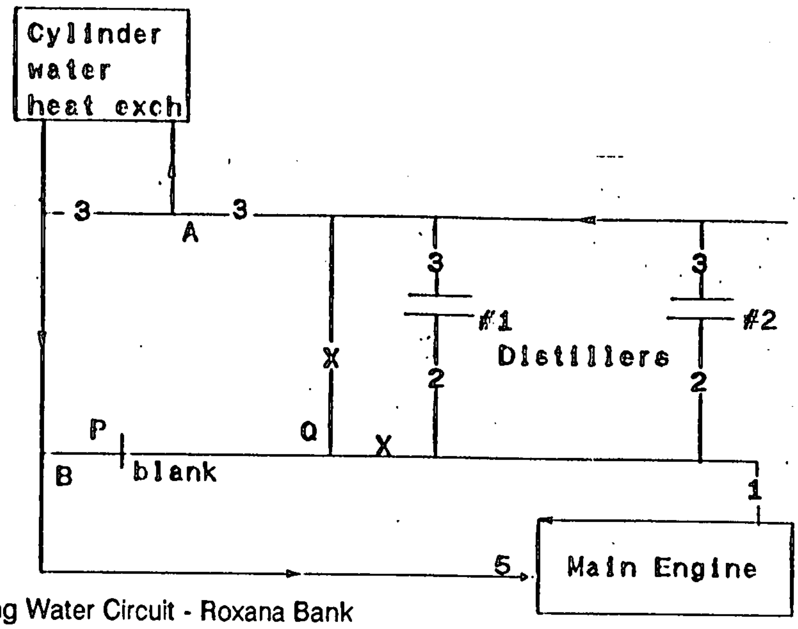


Figure 5.2a Cylinder Cooling Water Circuit - Roxana Bank

Table 5.3 Nomenclature for Figure 5.2a

point 1:	main engine outlet temperature
point 2:	FW distiller (#1, #2) inlet temperature
point 3:	FW distiller (#1, #2) outlet temperature, heat exchanger inlet temperature and bypass valve A temperature
point 4:	heat exchanger outlet temperature
point 5:	engine inlet temperature after mixing and ignoring pump work.

Operating conditions have full bore flow of the cylinder cooling water through the #1 FW distiller (84 m^3/hour) resulting in a lower temperature differential (large flow volume - low temperature differential). The incumbent engineers favour this practice and have also installed a blank in pipe PQ to maintain full flow conditions at temperature control valve A. Temperature control is fully manual and is therefore subject to operator error as can be seen by the varying heat exchanger temperature inlet (#3) and the engine temperature inlets (#5) when the load changes

(See Table 5.2 Engine operating temperatures -Roxana Bank). The resultant disadvantage of this phenomenon causes an erratic main engine outlet temperature.

Bypass temperature control valve B is at present blanked off as indicated in Figure 5.2a Cylinder cooling water circuit - Roxana Bank. The purpose of valves A and B are to prevent undercooling of the main engine and to ensure that engine operating parameters maintain a steady state value.

Valve B is required more in colder sea water temperatures. Pump and heat exchanger configuration as shown in Figure 5.1 Existing sea water cooling system - Roxana Bank, full bore flow exists at all times regardless of sea water temperature, requiring the use of valve B at colder temperatures.

The heat balance and heat loss (dissipated) of the existing circuit is shown in Table 5.4 Heat load cylinder water cooling and using the formula:

$$Q = MC_p(T_i - T_o) \qquad \text{-- (3-20)}$$

and temperatures monitored during the period of maximum engine load on the 09/10 November 1989.

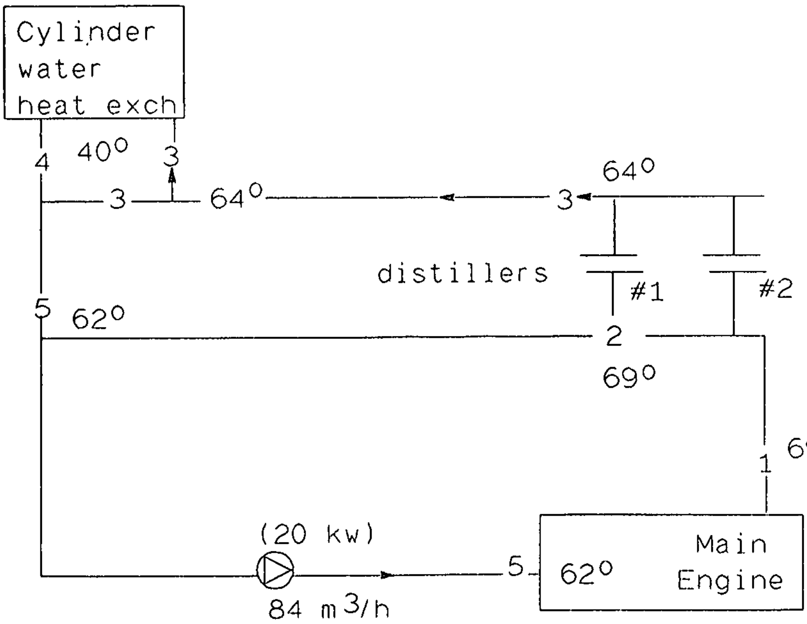


Figure 5.2b Cylinder cooling water circuit with temperatures - Roxana Bank:

Table 5.4 Heat Load - Cylinder Water Cooling - Roxana Bank

#1	#2	#3	#4	#5	Q_{dist}	Q_{THE}	Q_{eng1}	Q_{eng2}	$Q_{(sw)}$
°C	°C	°C	°C	°C	kW	kW	kW	kW	kW
69	69	64	40	62	479,4	192,8	692,2	703,3	693,2

From calculation {(a-2 Appendix B.5.2}, the volume flow through the cylinder water heat exchanger is 7 m^3/hour as there has already been a large amount of heat recovered in the FW distiller.

The low volume flow results in the heat exchanger being grossly underutilised and is extremely inefficient in operation. As the jacket water is the minimum fluid (in other words, the controlling fluid), calculation will prove that the sea water outlet from the heat exchanger will be 40°C. Calculation of the sea water heat absorbed using a differential temperature of 7° verifies this assumption. The heat exchanger inefficiency is due to the low volume flow and hence mass flow through the shell. With a shell diameter of 800 mm and an equivalent diameter D_e of 270 mm the resultant fluid velocity will be minimal as shown when using the equation for mass flow calculation:

$$m = \rho A V$$

where m = mass flow in kg/s

ρ = fluid density in kg/m^3

A = flow area in m^2

V = the fluid velocity in m/s

The Reynolds number will still remain in the laminar region regardless of the effect of the internal baffles. The low R_e inhibits heat transfer due to a low heat transfer coefficient which becomes the controlling coefficient in equation (3-5).

The present method of operation results in a system pressure head equivalent of 280 kPa. In this case the system losses are:

(1) pressure drop - main engine

(2) pressure drop - #1 FW distiller

(3) pressure drop - cylinder water heat exchanger

(4) Valve losses (9)

(5) pipe length - total length 20 m equivalent head loss

(6) reducing diameter

(7) increasing diameter

(8) bends - 20 - equivalent head loss

(9) discharge head - approximately 4 m

All piping is unlagged and therefore there is some heat loss through the conduction to the surrounding atmosphere. However, due to the lower temperature differential (ambient 30°C) and short lengths of piping, these losses will be considered negligible and are therefore ignored.

The results shown in Table 5.4 shows a total of 692,2 kW with 192,8 remaining available for heat recovery.

Combining the projected feed water volume of 10 m^3/hour with this remaining heat load will result in a feed water temperature rise of 16,7°C as shown. The feed water temperature drawn from the hotwell is 40°C. Thus with reference conditions at 40°C:

$$\rho = 994 \text{ kg/m}^3$$

$$C_p = 4{,}1781 \text{ kJ/kgK}$$

(Source: Appendix D.1.4)

Therefore $$Q_{fw} = \{10 * 994{,}6 * 4{,}1781 * (T_o - 40°)/3600$$

$$T_o = 56{,}7°C$$

Should this heat be recovered after the FW distiller, the jacket water outlet temperature would be in the order of: reference conditions at 64°C:

$$= \quad 983{,}2 \text{ kg/m}^3$$

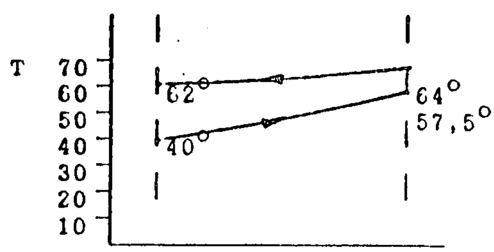
$$C_p \quad = \quad 4{,}1827 \text{ kJ/kgK}$$

(Source: Appendix D.1.4)

Therefore $\quad 192{,}8 \quad = \quad \{84 * 983{,}2 * 4{,}1827 * (64 - T_o)\}/3600$

$$T_o \quad = \quad 62°C$$

The temperature program will give a LMTD as shown in figure 5.3 Temperature program heat recovery 192,8 kW.



$\theta 1 = 22°$

$\theta 2 = 7{,}6°$

$LMTD = (22 - 7)/\ln 22/7$

$= 12{,}57°$

Figure 5.3 Temperature Program - Heat Recovery 1982,8 kW

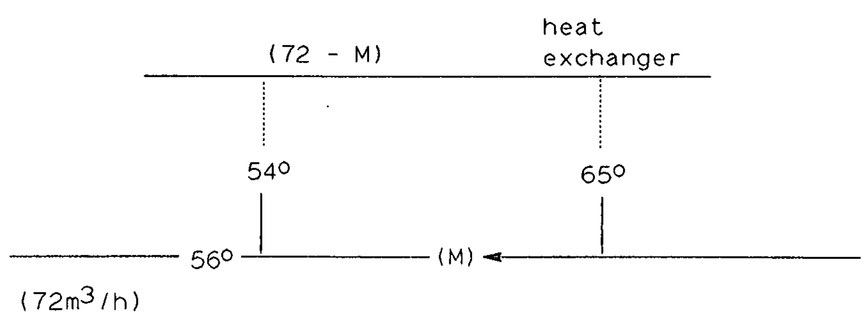
5.4 Lubricating Oil System

The lubricating oil system consists of the main circulating pump, the main engine and heat exchanger with the temperature program at 60% engine load as shown in Table 5.3 L.O. temperature program.

Table 5.5 Lubricating Oil Temperature Program - Roxana Bank

Lubricating Oil						Sea Water				
HE In	He out	Eng In	Eng out	V		HE In	HE out	V		Q
°C	°C	°C	°C	m^3/h	km^3	°C	°C	m^3/h	km^3	kW
65	54	56	65	72	880	28,4	31,5	87	1031	317

Calculations on the temperature program In Table 5.5 L O temperature program - Roxana Bank shows that there Is a quantity of lubricating oil bypassing the heat exchanger. The mass flows are thus:



Therefore (72 * 56) = {(72 - M) * 54} + (M x 65)

M = <u>13,1 m^3/hour</u>

The heat load specified in (Appendix C.6.4) based on an ideal temperature program for the type of engine fitted on the Roxana Bank is 66,4°C inlet and 55°C outlet from the heat exchanger. Assuming full flow of the lubricating oil through the heat exchanger due to a higher inlet cooling water temperature and the same outlet and inlet temperatures of 65° and 56°C, the heat load will still be 317 kW.

In other words: $Q = MC_p\Delta_t$

reference 60,5°C $\rho = 880\ kg/m^3$

$C_p = 2{,}009\ kJ/kgK$

(Source: Appendix D.1.4)

therefore	Q	=	{72 * 880 * 2,009 * (65 - 56)}/3600
	Q	=	317 kW

Confirming the heat balance on the cold fluid side:

	Q	=	$M_{sw}C_{psw}\Delta_{tsw}$
reference 30°C	ρ	=	1030 kg/m^3
	C_p	=	3,95 kJ/kgK

(Source: Appendix D.1.4)

therefore	Q	=	{87 * 1030 * 3,95 * (31,5 - 28,5)}/3600
	Q	=	<u>307,7kW</u>

The discrepancy of 3,3 kW can be attributed to local thermometers with small scale intervals and pump power inputs.

With the arrangement as fitted on the Roxana Bank, the lubricating oil heat exchanger and charge air cooler are fitted in parallel with the inlet temperature to the charge air cooler being 18°C. The heat load of the charge air cooler is again found from:

	Q	=	$M_{sw}C_{tsw}\Delta_{csw}$
reference 21,5°C	ρ	=	1032 kg/m^3
	C_p	=	3,95 kJ/kgK

(Source: Appendix A.4.9)

	Q	=	{87 * 1032 * 3,95 x (25 - 18)}/3600
	Q	=	<u>689,6 kW</u>

The LT circuit being fresh water at an inlet temperature of 32°C will yield a total heat dissipation value with the lubricating oil heat exchanger and the charge air cooler in series:

(1)	Q	=	$M_{fw}C_{pfw}\Delta_{tfw}$
reference 32°C	ρ	=	996,9kg/m^3
	C_p	=	4,1785 kJ/kgK
	307,7	=	{80 * 996,9 * 4,1785 * (To - 32)}/3600
	T_o	=	<u>35,3°C</u>

The temperature To is the inlet temperature into the charge air cooler giving an outlet temperature of:

(2)	Q	=	$M_{fw}C_{pfw}\Delta_{tfw}$
reference 35,3°C	ρ	=	995,6 kg/m^3
	C_p	=	4,1782 kJ/kgK
			(Source: Appendix D.1.4)
	689,6	=	{80 * 995,6 * 4,1782 * (To - 35,3)}/3600
	T_o	=	<u>44,3°C</u>

Thus the volume flow of 80 m^3/hour will have a temperature of 44,3°C at inlet to the central cooler.

5.5 Auxiliary Engines and Electrical Power Sources

There are two auxiliary engines type Sulzer 5AL 25 coupled to two 630 kVa 50 hz alternators. The engines run on gas oil only. In addition to these engines, there are two 700 kVa alternators driven off the main engine gearbox which can be clutched in or out depending on load requirements.

The electrical load as specified in the builder specification and requirements (Source; Fishing Trawler(b))

5.5.1 vessel steaming - 490 kVa

5.5.2 vessel fishing - 1045,8 kVa for fishmeal plant, refrigeration plant and general use. One 700 kVa alternator is used for the trawl winch.

At no time during the monitoring period on board the Roxana Bank on the 09/10 November 1989 did the load reach the theoretical values as laid down in the specifications. In fact the maximum load on one 700 kVa Power Take Off alternator was 390 kW. The second 700 kVa machine is kept coupled and on standby for the trawl winch, which has an intermittent operation. The actual load is therefore very much reduced in relation to the projected specified load. The owners have carried out a conversion to enable all the load to be carried by the 2 x 700 kVa units and as can be seen from the fuelling return sheets (Appendix C.6.5), the use of the auxiliaries have been phased out and resultant saving in gas oil can be seen in the gas oil consumption figures. The auxiliary engines are now only used in port when the main engine is shut down and when the main engine is stopped at sea. Referring to the fuelling return sheets, it can be seen that the auxiliary engines see very little service.

5.6 Auxiliary Boiler

The auxiliary boiler is a combination boiler consisting of two heat transfer sections:

5.6.1 The lower section which is oil fired, giving the boiler an output of 2500 kg/hour. The burner, now converted to operate on blended fuel consumes approximately 200 kg of fuel per hour at maximum output. The burner will modulate to maintain the set operating steam pressure of 700 kPa (gauge).

The boiler is a water tube type with a cross flow heat exchange operation.

5.6.2 The upper section which uses heat extracted from the main engine exhaust gas to generate steam. The output from test and trial data is 800 kg/hour. There is an induced draft fan situated in the exhaust trunk to assist the gas flow. (Capacity 7000 m^3/h at 25 mm Wg). This section is a water tube type and the exhaust gas passes over the tubes in a crossflow configuration.

The individual uptakes remain separate with no connection. The engineers on board operate the exhaust gas bypass flap at 50% - the reason being that it prevents overpressure when the vessel is operating at a reasonable load and the steam demand is minimal due to the fish meal plant being shut down.

During fish meal plant operation the burner was in operation at all times with a projected throughput of 134 kg/hour of fuel, giving a consumption of 2616 kg/24 hours.
During the period of idling, there was sufficient steam generated by the exhaust gas to supply the #2 FW distiller when the fish meal plant was shut down.

The exhaust gas boiler thus has a total output of 3300 kg/hr (maximum) at a nominal 700 kPa (gauge). At present on the Roxana Bank, the steam is used in the fish meal plant and #2 FW distiller.

5.7 Fresh Water Distilling Plant

The vacuum flash type distillers are designed to produce fresh water from sea water, and the two units use the energy from the main engine cylinder cooling outlet water or from saturated steam from the boiler. The output is 2 x 12,5 tonne/24 hours and the primary technical characteristics are:

5.7.1	shell pressure	3 - 0 kPa
5.7.2	cylinder water flow (heating)	55 m^3/hour
5.7.3	saturated steam demand	0,162 kg/sec
5.7.4	cylinder water temp. (heating)	60° - 65°C
5.7.5	saturated steam pressure	350 - 600 kPa
5.7.6	heat requirement	442 kW
5.7.7	cooling water requirement	75 m^3/hour

The distillers can operate at higher temperatures, the same heat load being transferred to the sea water. A benefit could be reduced volume flow, but care must be taken not to undercool the engine cylinder water. The HTHRU will be fitted before the FW distillers in order to reduce the heating water inlet temperature. It is important to note that a fresh water distiller using cylinder water as a heating medium is a well recognised form of heat recovery.

The engineers operating the distiller plant on the Roxana Bank have the full cylinder water flow of 84 m^3/hour flowing through the heating section of #1 FW distiller, before passing through the bypass valve and heat exchanger. With this full flow and temperature drop of approximately 4,5°C, the heat load transferred to the feed water is:

$$Q = M_{fw} C_{pfw} \Delta_{tfw}$$

reference 70°C ρ = 979,8 kg/m^3

C_p = 4,1876 kJ/kgK

(Source: Appendix D.1.4)

$$Q = (84 * 979{,}8 * 4{,}1876 \times 4{,}5)/3600$$

$$Q = \underline{430\ \mathrm{kW}}$$

which confirms the value of 424 kW as laid down in the specifications.

The engineers on the Roxana Bank have also found that the operating steam pressure of 280 kPa is sufficient for good water production.

5.8 Fish Meal Plant

The principal components of the fish meal plant on board the Roxana Bank are:

5.8.1 the raw product hopper and cooker

5.8.2 the motorised centrifugal liquid extractor

5.8.3 the steam drier operating at 80°C

5.8.4 the high speed transport blower, secondary drier fans and conveyor

The fish meal plant operates during the factory operation and is dependent on it. During the monitoring period on board the Roxana Bank on the 09/10 November 1989, the catches were relatively small and the factory did not have a 24 hour operating period. Extracts from the Roxana Bank's official log books and factory manager's operating log, show that a virtually 24 hour operating period does occur when the catch is large.

Examples of this are:

17 October 1989 -

Factory shut down	02h00,
Factory start	09h00, this operated carried through to 06h00 on 18 October 1989

20 October 1989

Factory shut down	02h30
Factory start	09h00, this operation carried through to 02h00 on 21 October 1989

In both cases the fish meal plant operated for 24 hours.

The fish meal plant processes the offcuts, offal and bycatch of the trawl and is capable of converting 50 - 60 tonnes of raw fish to 8 - 12 tonne of fish meal per 24 hours.

The heating steam consumption is 0,333 - 0,36 kg/sec and the operating pressure is 400 - 600 kPa.

As can be seen from the boiler output and the fish meal plant steam requirements, the burner would have to modulate to maintain steady steam production. If the burner modulation can be reduced there will be saving in fuel consumed.

Electrical consumption is 113 kW.

5.9 Fuel System

The Roxana Bank's main engine can operate on blended fuel with principal analysis as follows (Appendix B.5.1)

5.9.1.1	density @ 20°C	0,9628	
5.9.1.2	viscosity @ 50°C	66,28	cSt
5.9.1.3	flash point	73°C	
5.9.1.4	percentage marine fuel oil	87,4%	
5.9.1.5	percentage marine gas oil	12,6%	

The fuel is held in double bottoms and is pumped to a settling tank from where it is purified before use in the main engine.

Tank capacities are:

(1)	Blended fuel oil	790,4 m^3
(2)	Gas oil	261,5 m^3
(3)	Settling tank	24,5 m^3
(4)	Service tank	10,5 m^3

The fuel to the main engine requires heating to maintain the correct viscosity at the fuel pump and fuel valve. The fuel heating is accomplished as follows:

5.9.2.1 An outflow heater fitted in the service tank suction - electrical consumption 27 kW.

5.9.2.2 An inline heater fitted to the fuel supply of the main engine - electrical consumption 27 kW

5.9.2.3 A purifier heater - electrical consumption 21 kW

The fuel oil temperatures monitored on the Roxana Bank during the period 09/10 November 1989 are:

5.9.3.1	fuel to engine	95°C
5.9.3.2	service tank outlet	92°C
5.9.3.3	fuel purifier temperature	70°C
5.9.3.4	lubricating oil purifier temperature	72°C

The fuel to engine, service tank and fuel oil purifier are electric heaters as mentioned, with the lubricating oil purifier heater being steam heated. The temperature rise across this steam heater is 15°C.

In order to reduce electrical load on the alternators, all or partial heating of the fuel should be done by using steam. The heat recovery system design in Chapter Four has designed a fuel heating system using excess feed water from the HTHRU. A problem will exist though when the load on the main engine is reduced due to the reduction in exhaust gas mass flow rate, which in turn will have an effect on the exhaust gas boiler steam output.

5.10 Refrigerating Plant

The suitability of the refrigerating plant condensers has been discussed in Chapter Four, Section 4.5 Refrigerating plant condensers, and with a temperature rise of 2°C, they are not suitable for heat recovery.

The plant specifications are:

5.10.1 three screw compressors of 647,9 MJ/hour and one screw compressor of 2466 MJ/hour

5.10.2 three sea water circulating pumps operating in parallel {(volume flow x 3) constant head}, with a total output of 264 m^3/hour

5.10.3 four condensers of equal size which will receive a volume flow of approximately 66 m^3/hour when all four compressors are in operation.

The frozen fish storage can maintain 1092 tonne of fish at -30°C. The hold #3 is kept for the storage of fish meal.

Generally two compressors run in series with the third machine in a low stage to high stage configuration and supply refrigerant to the five plate freezers and the fish holds. The high temperature stage supplies the condensers and the circuits. There is a refrigerant heat exchanger in which the returning cold gas is heated by the compressor discharge gas resulting in higher temperature and liquid free suction conditions. The plate freezers and fish hold circuits are run continuously and the compressor loading is 80 - 100%.

Assuming a volume flow of 66 m^3/hour to the condensers and using the temperature rise monitored on the Roxana Bank on the 09/10 November 1989, the heat load can be calculated for each condenser.

Thus using the equation:

$$Q = M_{sw} C_{psw} \Delta_{tsw}$$

reference 20°C $\rho = 1032\ kg/m^3$

$$C_p = 3{,}98\ kJ/kg/K$$

$$Q = (66 * 1032 * 3{,}98 * 2)/3600$$

$$Q = 150{,}6\ kW \text{ per condenser}$$

5.11 Vessel Operations

The Roxana Bank carries out the actual trawling operation during daylight hours, the fishing commencing at sunrise. A typical days operations consists of the following program. Days 17 October 1989 and 20 October 1989 are lifted from the Roxana Bank's official log book and day 09 November is from the monitoring period.

Thus:	00h00 - 06h00: steaming
day	06h00 - 09h00: trawling
17.10.89	09h00 - 10h30: steaming - returning to starting point
	10h30 - 13h30: trawling
	13h30 - 14h15: steaming - returning to starting point
	14h15 - 19h00: trawling

	19h00 - 24h00: drifting
day	00h00 - 04h00: drifting, main engine off - repairs
20/10/89	04h00 - 06h45: steaming to fishing grounds
	06h45 - 09h00: trawling
	09h00 - 98h45: steaming - returning to starting point
	09h45 - 12h00: trawling
	12h00 - 12h45: steaming - returning to starting point
	12h45 - 14h45: trawling
	14h45 - 16h00: steaming - returning to starting point
	16h00 - 19h00: trawling
	19h00 - 24h00: steaming to new fishing sector
day	00h00 - 04h50: drifting - main engine idling
19/11/89	04h50 - 08h15: trawling
	08h15 - 09h30: steaming - returning to starting point
	09h30 - 14h15: trawling
	14h15 - 15h30: steaming - returning to starting point
	15h30 - 19h35: trawling
	19h35 - 22h00: steaming to new fishing sector
	22h00 - 24h00: drifting - main engine idling

As previously mentioned the fish meal plant operation is dependent on the factory operation which can be intermittent as on 09 November 1989 or full time as mentioned in section 5.8 Fish meal plant.

Thus, the whole process operation of factory and fish meal plant is totally dependent on the size of each catch and this can vary quite considerably from day to day and area to area. Days 17 October 1989 and 20 October 1989 were exceptional days with regard to catch and fish processed for storage on this particular voyage up to the monitoring period.

The main engine operates at 500 rev/min at all times as opposed to 530 rev/min as specified in the Sulzer bulletin 4-107.059.310. (Appendix C.6.4) Load changes as the propeller pitch alters or the electrical load changes. As the Roxana Bank is required to maintain a steady towing speed, the propeller pitch control can be adjusted from the wheelhouse.

The actual operating programs with regard to temperatures vary considerably from the recommended (or ideal) operating conditions. These discrepancies will be discussed in chapter six.

The hotwell temperature remained steady at 40°C resulting in all steam returns (or condensate) having the maximum heat extracted.

5.12 Summary

5.12.1 The heat exchangers on the Roxana Bank are dependent on the prevailing sea temperature which will render the lubricating oil and the cylinder water heat exchangers inefficient.

5.12.2 The engine operating temperatures vary depending on the particular operation. The FW distiller has recovered a large portion of the available heat from the cylinder cooling water.

5.12.3 At present operating temperatures there is no usable heat available from the lubricating oil circuit and the majority of the cylinder water heat load is recovered.

5.12.4 The boiler burner operates on two nozzles during the fish meal plant operation due to the high steam demand.

5.12.5 The electrical load is less than the projected specification load and all this load is carried by the shaft power take offs, resulting in a considerable saving in gas oil as the auxiliary engines are not used.

5.12.6 The vessel operations on a typical day show that the main engine load can vary from 20% to 60% of maximum continuous rating (MCR).

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CHAPTER SIX

6.1 Overview

The various methods of heat recovery are discussed as to the suitability and the design of a high temperature and low temperature has become of paramount importance whether a heat recovery system is installed or not due to the fact that fresh water central cooling are cost effective.

For ideal operating and steady load conditions, the heat recovery system will operate although there are limitations to this system.

The actual operating conditions show that due to the fluctuating engine loads, the projected heat recovery has severe limitations placed on it and the assumptions made for the turbine operations are not possible due to the lack of driving steam generated by the main engine exhaust gas. The heat load fluctuations are shown illustrating the difficulty in effective heat recovery and the increase in engine operating temperature will not solve the problems.

It is shown that for effective heat recovery, greater engine output powers are required with greater fluid and mass flow rates, and the inlet temperature to the first heat recovery unit is very critical to successful heat recovery.

The discussion concludes that due to unfavourable fluctuations in operating conditions in the machinery in the Roxana Bank, a heat recovery system is not recommended, although a conversion of the cooling water system to a fresh water high and low temperature circuit with central cooling is recommended.

6.2 Discussion

The purpose of the study is to identify the waste heat sources in the Roxana Bank and convert these losses into a heat recovery system, therefore increasing the temperature of the feed water to the auxiliary boiler.

Furthermore, the suitability of the modification must be evaluated as applied to the Roxana Bank coupled with the suitability of the cost aspect and availability of heat exchangers and associated equipment for the modification process.

To reduce operating costs in any vessel, the diesel engine manufacturers and ship owners are concentrating on the four following directions: (Source: Gallois: 1981:61)

6.2.1.1 decreasing specific fuel consumption

6.2.1.2 enabling engines to use worse fuels

6.2.1.3 extending part load capacity

6.2.1.4 using as much waste heat as possible

If the heat calorific value generated by the consumption of fuel in a main diesel engine is taken as 100%, the energy which is effectively employed to propel the ship is only 32% as indicated in figure 6.1 Sankey diagram showing energy losses.

Consulting the diagram, it can be seen that much of the remaining energy is lost as exhaust heat loss, cooling loss and propeller loss. In order to contribute to energy saving in the Roxana Bank, effective use must be made of these heat losses to improve the efficiency of the propulsion engine and hence the overall propulsive efficiency. Improving the efficiency of the propeller is not covered in this study.

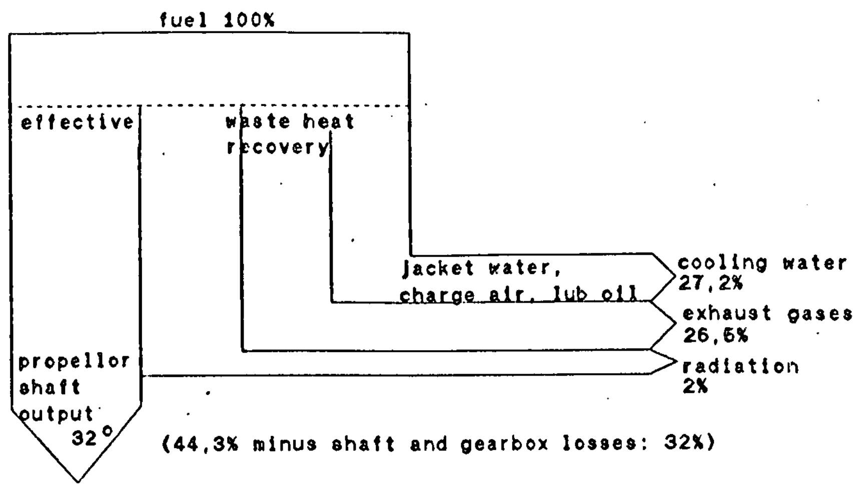


Figure 6.1 Sankey Diagram Showing Heat Losses

(Source: Morton TD1970:6)

Generally, the heat losses in the Roxana Bank main engine will be found in the exhaust trunk, cylinder cooling water, lubricating oil and charge air heat exchangers with the main refrigeration condensers also a consideration.

Section 6.2.1.4 only forms part of this study and with waste heat utilisation being successful, section 6.2.1.1 will be directly affected with a reduced specific fuel consumption as more work is gained from the input.

Referring to Figure 6.1 Sankey diagram showing energy losses; with the average engine output (brake) at approximately 40% before the gearbox, shafting and propeller losses, the other losses are 58% to exhaust and cooling water with a small percentage to irretrievable radiation losses of 2%. Part of the heat losses are utilised in the exhaust gas to drive the turbocharger making the engine more efficient at 39%. The cooling water losses including lubricating oil, charge air cooler and cylinder water approach 27% leaving a loss of approximately 26,5% to the exhaust.

With the higher thermal efficiencies found in diesel engines, more incentive has been created to exploit lower temperature waste heat sources, such as jacket cooling water, scavenge air cooling water and lubricating oil cooling water or directly from the scavenge air and lubricating oil. Thorpe and Armstrong (MER 1982:12) of the University of Newcastle on Tyne, in comparing fuel cost savings have calculated that scavenge air heat may be used for heating purposes in fuel oil bunker tanks, settling and service tanks, thus reducing the required steam for these services by approximately 34%. They also found that the utilisation of heat from the jacket cooling water was theoretically possible for bunker tank heating due to the lower temperatures and heat values but should preferably be kept for heating the FW distillers if fitted.

The majority of waste heat at this stage is of low quality and for efficient heat recovery to take place, the heat exchangers will be from the design consideration large due to low LMTD's and increased thermal length.

The majority of the capital cost is in the heat exchangers and this cost together with the saving in fuel must be correlated with the payback period to determine whether any conversion is financially feasible.

The design of a feed system using waste heat to increase the boiler feed water temperature will include heat recovery from the low temperature exhaust, jacket cooling water, lubricating oil and charge air coolers.

Initially, there were several methods investigated to extract heat from the Roxana Bank's cooling water systems. The heat recovery system must be kept simple and be easily maintained. Capital cost must be kept to a minimum to ensure a short payback time, as savings only become apparent after payback time is completed.

The methods under investigation were :

6.2.2.1 In an attempt to keep installation costs to a minimum, the conversion would consist of the existing plant layout remaining with the sea water system being converted to fresh water. A heat recovery unit would be installed in the circuit after mixing of the fluids after passing through the existing heat exchangers. (Appendix F - DMF002/89) A central cooler would still require fitting as the raw water system would be a closed circuit and must be cooled to 32°C. In this system, heat recovery would be at a minimum as shown in the calculations and was therefore discarded.

6.2.2.2 A second simple heat recovery system was investigated using a tank in the hull as a storage feed tank with the main engine circulating pump, the boiler feed pump and the refrigeration condenser pumps drawing water from this tank. The fluid would be fresh water throughout.

This system has many negative aspects being:

(1) Radiation losses to the sea through the hull would be excessive

(2) A large quantity of water would be required and thus would affect the Roxana Bank's stability and reduce the fuel carrying capacity

(3) Central coolers would be required in the main engine and the refrigeration plant circuits to ensure the inlet temperature of 32°C

(4) The heat recovery prospects are limited - maximum heat recovery is 47,2°C assuming infinite heat recovery

(5) A large amount of heat would have to be extracted and inlet temperature control would require complicated bypass valves

(6) Contamination of the feed tank from any source would result in entire system contamination and possible engine shut down causing hazardous conditions for the crew and the vessel. (Appendix F - DMF007/89)

Furthermore, investigations into heat recovery on the Roxana Bank suggested that other sources of heat recovery were available, namely, the large refrigerating plant which is in continuous operation. With the operation of two large freezer holds maintained at -21° and plate freezers to quick freeze the processed product, in continuous use, an extensive refrigerant compressor system is required. At all times there are at least three compressors in use with their associated refrigerant condensers.

During the monitoring period of the Roxana Bank's machinery on the 09/10 November 1989, the temperature rise across the sea water side of the condensers was 2°C. Using three pumps in parallel giving an estimated volume flow of 66 m^3/hour with an inlet temperature of 18°C.

The sources of heat loss on the Roxana Bank are therefore identified as follows:

6.2.3.1 high temperature exhaust gas (after turbocharger)

6.2.3.2 low temperature exhaust gas (after exhaust gas boiler)

6.2.3.3 jacket cooling water circulating water (main engine)

6.2.3.4 circulating lubricating oil (main engine)

6.2.3.5 charge air (main engine)

A further heat loss found on crosshead type engines with water cooled pistons will be from the piston circulating cooling water heat exchanger. (Appendix C.6.1)
Raising the feed water temperature will reduce the amount of sensible heat required to raise steam in the exhaust gas boiler. As monitored, the feed water temperature is 40°C and increases in this temperature with savings are illustrated on the Rankine cycle T-h diagram

Figure 6.2 Enthalpies of boiler feed water for temperatures from 40°C to 150°C based on an operating pressure of 800 kPa (absolute).

From the enthalpies shown and referring to Table 6.1 Temperature - Enthalpy the enthalpy savings are shown.

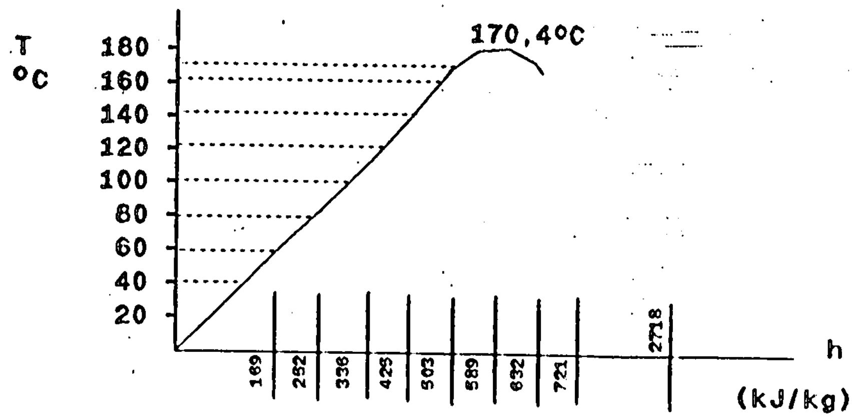


Figure 6.2 Enthalpies of Boiler Feed Water

Table 6.1 Temperature - Enthalpy

temp	enth (h_f)	h_g 800	save	reqd
°C	kJ/kg	kJ/kg	kJ/kg	kJ/kg
40	169	2768	-	2599
60	252	2768	83	2516
80	336	2768	84	2432
100	425	2768	89	2343
120	505	2768	80	2263
140	589	2768	84	2179
150	632	2768	43	2047
170	521	2768	89	2047
Total saving up to 150°C				436

The heat required per kilogram of steam reduces with increased feed water temperature. In a composite boiler with automatic control, this reduction can therefore that the feed temperature isto be as high as possible.

When calculating the heat transfer area of the HTHRU, the initial cost of construction will be of prime concern as this will determine the payback period and efficiency.

Referring to the graph and tables in Appendix C.6.2, C.6.2a and C.6.2b shows that for a decreasing pinch point (the difference between the cylinder water outlet and the feed water outlet), the heat recovered increases. However, the reducing pinch point shows a dramatic increase in heating surface area.

During computer runs on a hypothetical plant fitted with Sulzer RND90M engine, the reducing pinch point in the case of an exhaust gas boiler showed large increases in heating surface area for small increases in heat recovered at the upper limits (Anon 1978b:69).

The calculated HTHRU temperature program raises the feed water temperature to 73°C only. The program has been selected as the ideal main engine jacket outlet temperature is 85°C and coupled with an inlet of 75,8°C, gives a heat load of 896 kilowatts.

The requirements of the FW distiller is 440 kW, placing a further heat recovery of 450 kW which limits the HTHRU outlet temperature to 73°C.

A feed water temperature of 73°C, while an improvement on 40°C is clearly insufficient to bring about any substantial savings in enthalpy and therefore a significant reduction in fuel consumed. To overcome this low feed temperature, two methods can be applied to the Roxana Bank:

6.2.4.1 utilise an economiser in the low temperature exhaust gas trunk - after the exhaust gas boiler

6.2.4.2 raise the jacket cooling water temperature to 120°C

6.2.4.1 Ideally, to maintain the stack temperature at 180°C to prevent acid attack, an economiser could be fitted after the boiler as the exhaust temperature after the boiler is 300°C at 60% load on the Roxana Bank giving a temperature differential of 120°C and be utilised to raise the feed water to acceptable temperatures.

The feed water must bypass the economiser during low load conditions to prevent undercooling of the exhaust gas and promote acid attack. The bypass system would require to be automatic and would affect the whole plant if fluctuations occur. Utilising the economiser to preheat the total mass flow (for example, 2,7 kg/sec) would make available a higher temperature fluid in the excess feed water circuit for greater heat recovery.

However, problems would occur if the heat exchangers were placed out of service, resulting in a high temperature hotwell and an initially high temperature into the HTHRU.

The temperature program of an exhaust gas/feed water economiser, assuming a feed water flow of 10 m^3/hour and an exhaust outlet temperature of 180°C will yield:

$$Q_{ex} = M_{ex} C_{pex} \Delta_{tex}$$

$$= 3{,}69 * 1{,}016 * (300 - 180)$$

$$Q_{ex} = \underline{450 \text{ kW}}$$

See Appendix A.4.8 for calculation of exhaust gas specific heat value C_{pex})

The mass flow of exhaust gas of 3,69 kg/s is the assumed quantity calculated at 60% load.

Thus the temperature rise of feed water:

reference 73°C = 960,6 kg/m^3

C_p = 4,2133 kJ/kgK

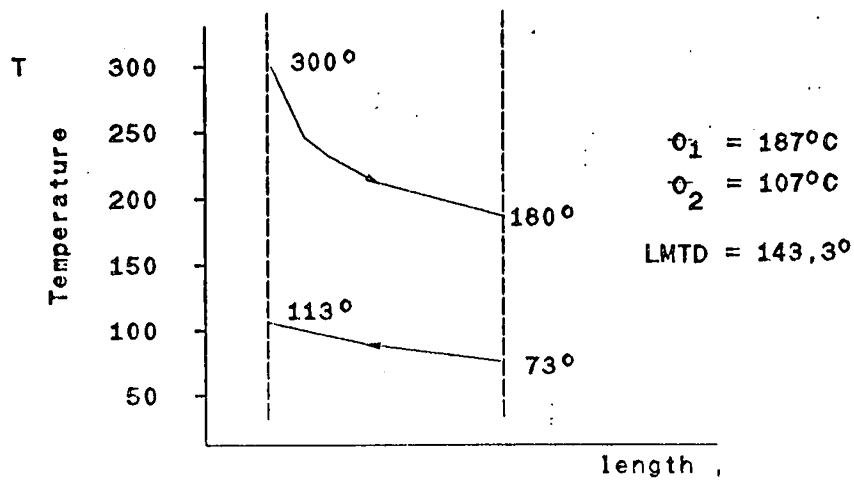
(Source: Appendix D.1.4)

$$Q = M_{fw}C_{pfw}\Delta_{tfw}$$

$$450 = \{10 * 960,6 * 4,2133 * (T_o - 73)\}/3600$$

$$= 113°C$$

resulting in a LMTD of



And a thermal length $\theta = (t_i - t_o)/LMTD$

$$= (300 - 180)/143,3$$

$$\theta = 0,84 \text{ (on the gas side)}$$

The θ value indicates that a relatively small heat exchanger will be required to raise the feed water to 113°C.

An alternative is to heat the boiler requirement of 0,69 kg/second which will result in an inlet temperature of 200°C. However at this temperature, the pressure for pumping must be 1550 kPa (absolute). This is far in excess of the 800 kPa (absolute) required pressure for the boiler operation on the Roxana Bank.

6.2.4.2 The second method which increases the jacket cooling water outlet temperature to 120°C will require a pump capable of maintaining the equivalent pressure of 300 kPa (absolute). As the monitored pressure on the Roxana Bank's main engine jacket cooling water system is 280 kPa (gauge), (see Table 6.2 Main engine temperatures, pressures and heat loads) no difficulties in this respect are anticipated.

The data collected during the monitoring period 09/10 November 1989 show a marked difference to the ideal conditions found in the Sulzer bulletin (Appendix C.6.4) and these with the shipyard trial data (Appendix C.6.3) are shown in Table 6.2 Main engine temperatures, pressures and heat loads.

The jacket cooling water heat load to be dissipated has a higher value than the bulletin value, but cannot be utilised on the Roxana Bank because of the lower engine outlet temperature of 68°C. With a temperature of 68°C assuming infinite heat exchanger length, the feed water would only reach 68°C.

Table 6.2 Main Engine Temperatures, Pressures and Heat Loads

	Jackets				Lubricating Oil				Charge Air			
	eng in	eng out	press	Q	eng in	eng out	press	Q	CW in	CW out	press	Q
	°C	°C	kPa	kW	°C	°C	kPa	kW	°C	°C	kPa	kW
bulletin	76	85	250	85	55	66	600	411	32	42,3	--	990
09 Nov 1989	60	68	280	1051	54	65	510	311	18	25	95	688
ship trial	58	65	300	--	47	55	540	283	--	--	170	640

The lower value of the heat load in the lubricating oil is due to the reduced flow through the heat exchanger as a result of the lower sea temperatures. The sea water is at full flow through this heat exchanger.

A prime source of heat recovery in the high temperature section of the intercooler is reduced as the heat dissipated is considerably lower. Taking into account that the scavenge air energy decreases by more than the square of the power at reduced engine loads, this particular source of waste energy is only feasible for exploitation in very high power installation. (Anon 1988b:45)

Furthermore, as the engine load fluctuates, it can be seen that the heat values for the different heat exchangers fluctuate as shown in Table 5.2 Engine operating temperatures -Roxana Bank.

Gallois (Anon 1978b) and Anon 1986 have stated that for efficient and sufficient steam generation from an exhaust gas boiler using the exhaust as the only heating medium to drive a steam turbine to provide electrical power, the feed water to the boiler should be at least at 120°C.

As there are limitations placed on increased steam generation prospects in the Roxana Bank as discussed in Section 6.3 Subproblem 2 - Assumptions and Results, the high feed temperature is not required, but fuel saving at the boiler burner would be desirable.

It can be seen that although the jacket cooling water temperature has been raised to 120°C and the temperature differential at 10°C, the heat load will remain at 946 kW.

A further source of heat recovery that was to be evaluated as to heat recovery prospects was the direct contact heater which operates with primary circulating water extracting sensible and latent heat from the engine exhaust as the water comes into direct contact with the exhaust gas through sprayer in the exhaust gas trunk. This primary water then transfers the heat gained to the secondary water (feed water) via a conventional heat exchanger.

A limitation placed on this type of heat recovery system is that due to the low exhaust gas temperature after the heat transfer process, the exhaust gas has reached a temperature well below that of the dew point and condensation takes place with the resultant formation of sulphuric acid. Thus exhaust gas trunk and upper levels of the boiler must be of corrosion resistant material. The water sprayer system, water weir, pump and associated piping would also have to be of corrosion resistant material resulting in a high construction cost.

To obtain full benefit from this type of heat exchanger, Goldstick and Thumann (1977:127) have determined that the entering primary water to the sprayers must be of a sufficiently low temperature to extract maximum benefit from such a system. Thus the feed water inlet temperature to the conventional heat exchanger should also be of a sufficiently low temperature. Expected operating figures quoted are well below the expected feed tank outlet temperature envisaged on the Roxana Bank and therefore the full effect of the direct contact between the primary water and the exhaust gas would be lost due to the expected feed water temperature from the hotwell of 40°C.

A further limitation placed on this type of heat exchanger is the space required to fit the unit. Machinery space on the Roxana Bank is at a premium and the direct contact heater would have to be fitted in the starboard funnel after the exhaust gas boiler. Due to the nature of the deck layout, the deck engine casing and the funnel is just large enough to contain the exhaust gas boiler. The fitting of the direct contact heater and any further steel construction containment casing would require recalculation of the Roxana Bank's stability criteria, and the stability could be seriously affected.

Goldstick and Thumann (1977:132) have quoted case histories of plant fitted with Direct Contact Heaters and predict that the temperature outlet of the secondary fluid (feed water) is an expected 43°C to 50°C.

To illustrate the unsuitability of a direct contact heater on the Roxana Bank, the conditions are taken at 60% engine power with the temperatures as monitored on the 09/10 November 1989, with the exhaust gas temperature of 300°C and an exhaust gas mass flow of 3,69 kg/sec (see calculation Pg 152). The assumption is made that the exhaust gas is to leave the funnel at 180°C.

Thus, the heat available from the exhaust gas:

$$Q = M_{eg}C_{pex}\Delta_{tex}$$

$$= 3{,}3{,}89 * 1{,}016 * (300 - 180)$$

$$Q = \underline{475{,}5\ kW}$$

Design conditions for a direct contact heat recovery unit require a ratio of liquid to gas flow rate (L/G) of 3 - 10 with the lower rate suitable for packed tower design and the higher rate for sprayer type. (Goldstick and Thumann 1977:145)

The sprayer water inlet temperature is assumed at 40°C which is the inlet to the heat exchanger from the feed water tank. Thus the rise in temperature of circulating water:

$$Q = M_wC_{pw}\Delta_{tw} \qquad \text{(select } L/G = 7\text{)}$$

$$475{,}8 = 7 * 3{,}89 * 4{,}1781 * (T_o - 40)$$

$$T_o = \underline{44{,}2°C}$$

With a temperature rise of 4,2°C and assuming the infinite length, the maximum water temperature rise would be 4,2°C. In reality, it would be lower. For the given temperature program the direct contact heater is not viable as a feed water heater as the temperature rise is too small and exhaust temperatures cannot be guaranteed to be steady at all times. Should the exhaust temperature differential be reduced to 50°C then the heat available becomes 938 kW. However, to achieve this, the unit would have to be manufactured from corrosion resistant material. Also, as the heat recovered increases so does the heat transfer surface area, resulting in increased capital cost.

The direct contact heater will for the reasons discussed, not be included at all in the Roxana Bank's heat recovery system.

The first subproblem requires the identification of energy losses on the Roxana Bank from the main engine cooling systems, the exhaust system and the refrigerating plant. As has been discussed, the low temperature direct contact heater and the refrigerating condensers have been discarded as these units are not compatible with the designed feed system envisaged for the Roxana Bank.

For the proposed feed system to be effective, the running conditions on the Roxana Bank must be steady with no fluctuations in mass flow rates or heat transfer rates. During the monitoring period on the Roxana Bank (09/10 November 1989), the engine data particulars recorded have been plotted on a heat load to time graph, illustrating the fluctuations found in the heat loads as the engine load fluctuated.

Figure 6.3 Cylinder water heat load shows the fluctuations in the heat load with the heat required for the FW distiller and remaining available heat for heat recovery. With the heat load altering as it does, the feed temperature would be erratic and would cause cycling of the automatic valves, the position of which are shown in figure 6.4 Proposed jacket water circulating system.

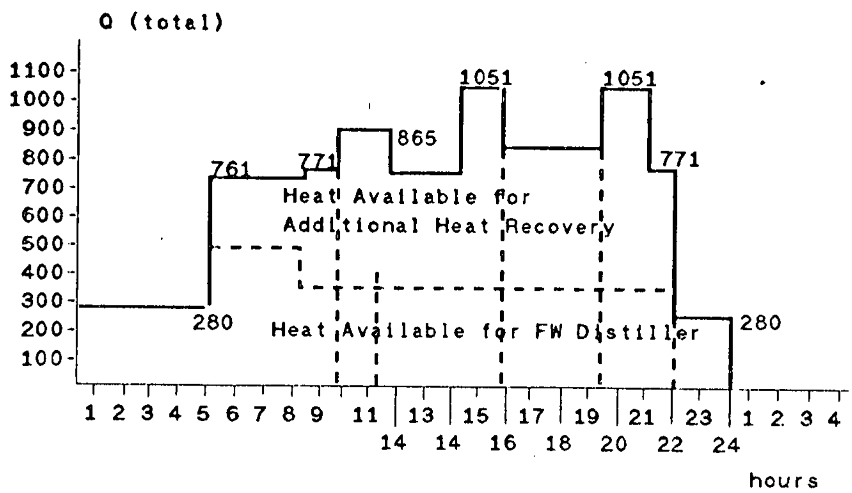


Figure 6.3 Cylinder Water Heat Load (Appendix B.5.3)

The HT thermostatic valve will be set at 85° with the LT thermostatic valve at 75°C. At an outlet temperature of 85°C from the main engine, there will be full flow through the HTHRU and no mixing of the fluids will take place at position A.

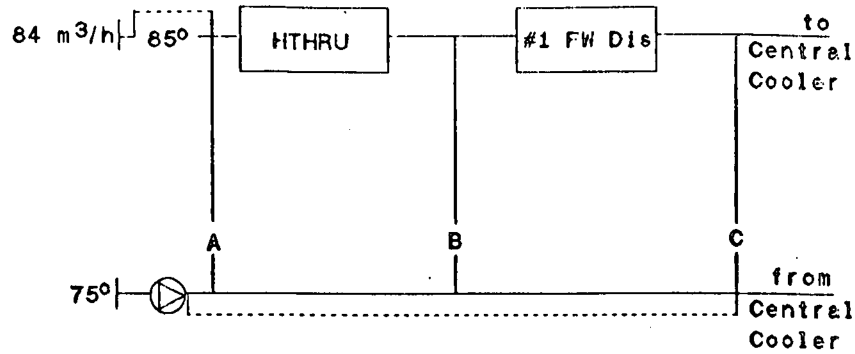


Figure 6.4 Proposed Jacket Water Circulating System

If the main engine cylinder water outlet temperature fluctuates due to reduced engine requirements as shown in figure 6.3 Cylinder water heat load, there will be reduced flow through the HTHRU and the FW distiller. A stage could be reached where little or no flow would pass through the two heat recovery units if the temperature differential reduces to approximately 2° under virtually no load conditions. Under operating conditions as found on the 09/10 November 1989, the heat available was all dissipated in the FW distiller, a value of 269 kW with no flow through the jacket cooling water heat exchanger. Referring to figure 6.4 Proposed jacket water circulating system, and with the heat recovery arrangements all available, heat would be dissipated by the HTHRU (an energy saving), but would require the fitting of an additional bypass control valve at position B, rendering the FW distiller out of service resulting in its shutdown. On the Roxana Bank, the FW distillers (#1 and #2) operate at all times regardless of engine load with the water production varying. To ensure that the FW distiller does not shut down, it should be installed before the HTHRU which reduces the available heat appreciably.

Table 6.3 Varying mass flows in jacket fresh water cooler shows the effect a reducing temperature (main engine outlet) has on the mass flow bypassing the HTHRU. The calculation is based on the assumption that for a fixed thermal length Δ of the heat exchanger, the primary fluid (jacket cooling water) at reduced mass flow will attain the secondary fluid outlet temperature and not be lower than this. In this application, the secondary fluid temperature program is 40,4° to 73°C.

Table 6.3 Varying Mass flows in Jacket Fresh Water Cooler

Temperature		Mass Flow	
main engine	HTHRU out	bypass	HTHRU
°C	°C	m^3/h	m^3/h
85	80,4	14	70
84	79,3	15,3	68,7
83	78,2	16,8	67,2
82	77	18,7	65,3
81	75,8	21	63
80	74,6	24	60
79	73,2	28	56
78	72,6	33,6	50,4
77	69,3	42	42
76	64,4	56	28
75	bypass	84	nil

(Source: Appendix B.5.3)

The calculated data in Table 6.3 Varying mass flows in jacket fresh water cooler, shows the mass flow variation assuming a fixed heat recovery of 368 kW and a theoretical primary outlet of 73°C. In practice this will not be so, as the heat load must be determined for each temperature change after the LMTD has been calculated by an iterative process that is both lengthy and cumbersome.

The calculation has been carried out to illustrate the effect a reducing main engine inlet temperature has on the mass flow available for the FW distiller.

The limitations of this kind of theoretical example will be found at an engine temperature outlet of 78°C resulting in a primary fluid outlet temperature of 72,8°C. Heat recovery of 368 kW is unobtainable at this temperature as the secondary fluid temperature could not reach 73°C.

The calculation proves however, the difficulty faced in placing the HTHRU and the FW distiller in series and the effect the reduced temperature has on the distiller. As has been borne out by operating conditions, the FW distiller will produce fresh water at low loads albeit at low production.

Ideally, the FW distiller should be placed before the HTHRU, but this will cause low feed water temperatures to the boiler.

From previous figures obtained from the vessel's logbook, it is possible to have the factory and fish meal plant operating for 24 hours, which will give a steady heat load for heat recovery purposes. Factory and fish meal plant operation is dependent on the size of the daily catch.

The Lubricating Oil Heat Exchanger

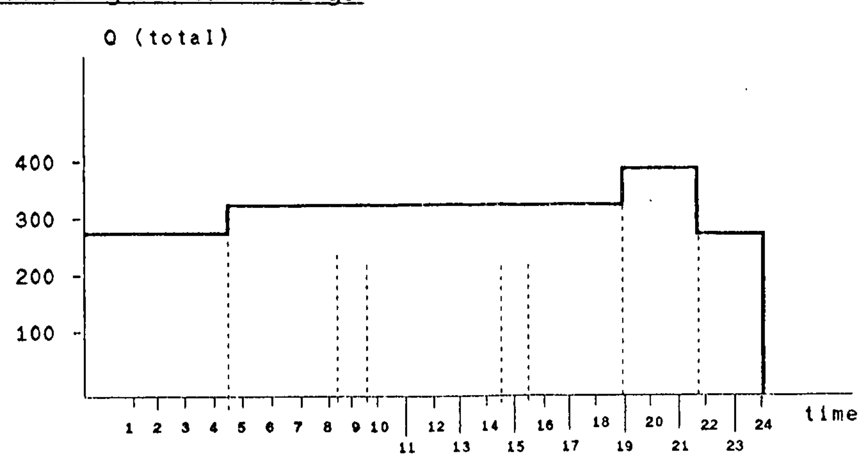


Figure 6.5 Lubricating Oil Heat Load

(Source Appendix B.5.5)

Referring to figure 6.6 Lubricating oil heat load - the value of 311 kW is constant from 04h50 to 19h30. The trawling commenced at 04h50 and was completed at 19h30. The vessel usually steams back to its original starting position after each trawl.

The heat load of the lubricating oil heat exchanger is more constant across the 24 hour period, but the heat load is low resulting in poor heat recovery prospects. Assuming full heat recovery with low mass flow and large thermal length, the maximum secondary fluid outlet temperature would be in the order of 65°C. The large thermal length however, would make the heat exchanger very expensive and produce unacceptably high pressure drops.

With the mass flows required for full heat dissipation, the temperature rise of the primary fluid (fresh water) can be found from the equation:

$$Q = M_{fw} C_{pfw} \Delta_{tfw}$$

using the maximum heat dissipation of 375 kW, and a fresh water inlet temperature of 32°C.

Assuming a large thermal length and low mass flow, to obtain a temperature outlet of 65°C:

reference 48,5°C

$$\rho = 990{,}67 \text{kg/m}^3$$

$$C_p = 4{,}1778 \text{ kJ/kgK}$$

(Source: Appendix D.1.4)

would require a mass flow of:

$$357 = \{M * 990{,}7 * 4{,}1778 * (65 - 32)\}/3600$$

$$M = \underline{9{,}4 \text{ kg/sec}}$$

or a volume flow of 34,2 m^3/hour which is not acceptable for heat recovery purposes.

The proposed system has the lubricating oil heat exchanger connected in series before the charge air cooler. The charge air cooler is another source of usable heat energy and there are systems using this heat source, particularly if the intercooler is split into high and low temperature sections. The heat recovered using this method increase dramatically (Figure 4.3 Available heat in charge air and jacket water:66) and is used as a feed water preheater in cases where maximum heat recovery is required for exhaust gas boiler feed water heating for maximum steam generation and turbine operation.

The Charge Air Heat Exchanger

Although the volume of air moved through the engine will remain constant, in other words:

$$Vol = (\pi/4) * D_c^2 * L_s * (N/60) * \text{no. of Cy1}$$

the mass flow will vary according to the charge air pressure and temperature. In this study, the characteristic gas equation:

$$PV = mRT$$

found from Boyle and Charles' Laws. R = 0,287 kJ/kgK for air. With the charge air cooling circuit connected in series with the lubricating oil cooler, the outlet primary fluid temperature can be found and using the maximum heat load of 788 kW, the outlet primary fluid temperature is:

$$Q = M_{fw}C_{pfw}\Delta_{tfw}$$

reference 35,9°C $\rho = 995{,}83 \text{ kg/m}^3$

$$C_p = 4{,}1783 \text{ kJ/kgK}$$

(Source: Appendix D.1.4)

thus $788 = \{80 * 995{,}8 * 4{,}1783 * (T_o - 35{,}9)\}/3600$

$$T_o = \underline{44{,}4°C}$$

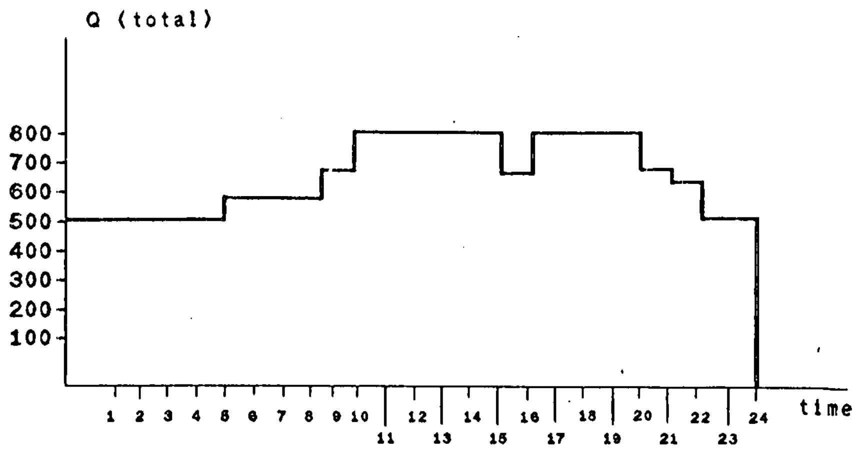


Figure 6.6 Charge Air Heat Exchanger Heat Load

Referring to the above results, it can be seen that the possible heat recovery from the actual cooling water (at 44,4°C) is of a very low quality and cannot be used in the application designed for the Roxana Bank. This heat can be used in vessels that operate in subzero conditions to preheat charge air before entry into the turbo compressor and for air conditioning. A further use for this fluid is to circulate it through fuel tanks to preheat the fuel oil prior to pumping either by through heating coils or thermopanels.

As this type of heating is by conduction, heating coils are large and can contribute a large portion of conversion costs. A further use for this low value heat is for feed heating although in this case the feed water must be of a sufficiently low temperature to recover heat from the LT circuit. In the case of a steam turbine installation operating under a vacuum exhaust, a temperature rise of 10°C could be expected. Further heating of the feed water would then come from the HT circuit.

As the LT water has no contribution to heat recovery as applied to the Roxana Bank, the full heat load from the LT system will be dissipated by the central cooler.

The temperatures recorded during actual operating conditions differ from the ideal conditions with regard to heat loads to the heat exchangers. With the sea water cooling at present at full flow and no temperature control on the sea water, the maximum heat to be dissipated by the lubricating oil heat exchanger fitted in the Roxana Bank is 375 kW. Under ideal operating conditions, the heat to be dissipated is 411 kW. The lower value is due to the reduced flow through the heat exchanger. The lower sea water temperature will have an effect on this reduced flow through the heat exchanger. With the higher inlet primary fluid temperature (raw water), the mass flow through the heat exchanger will increase resulting in a higher heat load.

The lubricating oil inlet temperature to the heat exchanger of 65° to 66°C will not change due to operating conditions. The temperature differential of 11,4°C is based on full load operating conditions with a full flow of 72 m^3/hour passing through the heat exchanger.

With the engine operating at 60% load, the temperature differential of 10° to 11°C can be expected with less flow through the heat exchanger. The lubricating oil thermostat will be set at 56°C inlet temperature to the engine.

Calculations show that the heat to be dissipated by the charge air cooler (maximum) is 788 kW. Again, the heat load of 990 kW is based on full load operation. With the LT water (raw water) leaving the lubricating oil heat exchanger at approximately 39°C, calculations based on the heat load of 788 kW will give an outlet temperature of approximately 44,5°C. However, the lubricating oil heat load and charge air cooler heat load fluctuate due to engine load fluctuation and therefore temperature fluctuation will occur.

Thus the minimum temperature rise through the LT section can be found from equation:

$$Q = MC_p\Delta_t$$

Thus with a minimum heat load of 276 kW the LT water exiting the lubricating oil cooler will be:

reference 32°C $\quad \rho = 996{,}9 \text{ kg/m}^3$

$C_p = 4{,}1785$ kJ/kgK

(Source: Appendix D.1.4)

$$276 = \{80 * 996{,}9 * 4{,}1785 * (T_o - 32)\}/3600$$

$$T_o = 35°C$$

thus the inlet to the charge air cooler will be 35° and the outlet is:

reference 35oC ρ = 996,1 kg/m^3

C_p = 4,1785 kJ/kgK

(Source: Appendix D.1.4)

$$496 = \{80 * 996{,}1 * 4{,}1783 * (T_o - 35)\}/3600$$

$$T_o = 40{,}4°C$$

During the prelimiary investigation undertaken while formulating the research proposal and the sources of energy losses in the Roxana Bank, several major assumptions were made. The assumptions were steady main engine loads which in turn would promote steady heat recovery. Some of these assumptions were:

(i) the fishing operation (trawling) was a 24 hour operation

(ii) the factory and fish meal plant operated for 24 hours because of the influence of (i)

(iii) the main engine operated at approximately 80% of maximum continuous rating during the 24 hour period

(iv) during peak load periods, the power take off alternators were hard pressed to supply the electrical load

(v) that sufficient steam would be available from the exhaust gas boiler only to drive a small turbo alternator to supplement the electrical load.

Assumption (i): Due to the dangerous nature of trawling operation, company regulations state that fishing operations may only take place during daylight hours. The monitoring of fishing times on the Roxana Bank as found on the 09/10 November 1989 show that the first trawl commences around 04h15 hours and the final trawl is completed at 19h00. Thus there is a period of approximately 9 hours where no trawling is done. If the Roxana Bank is steam between fishing areas, the steaming load is generally similar to the trawling load.

Assumption (ii) The limitation of the fishing periods will have a direct effect on the factory and fish meal plant operation. Although there are periods when the factory and fish meal plant are in 24 hour operation as shown on days 17 October and 20 October 1989 due to the large volume of fish caught, this phenomena is not constant; there are periods when the factory and fish meal plant are shut down as shown during the monitoring period.

Assumption (iii) The fish meal plant, factory and refrigerating plant have a large effect on the electrical load supply. Furthermore, the propellor blade pitch setting influences the engine load from the propulsion point of view. The resistance to motion applied by the trawl net when in the water to the Roxana Bank results in a lower pitch setting with the trawling speed at approdimxately 3,5 knots. Engine load is similar while the Roxana Bank is steam with process plant on, but a marked difference is visible when the process plant is off.

Based on the assumption that the engine coupled with the power take offs would operate at nearly maximum capacity on the Roxana Bank, the engine loading was assumed to be approximately 80% of maximum continuous rating (MCR).

In reality, the engine load on the Roxana Bank was in the region of 60% of MCR which has a maximum during the monitoring period. The 60% load will mean the reduction of exhaust gas mass flow in relation to gas mass flow at 80% with the attendant reduction in heat recovery, and less heat available for heat recovery purposes.

The engine runs continuously for the 24 hours as can be seen from the fuelling return sheets. The engine remains on blended fuel for the full period even at low operating loads - in other words, at 20% (Appendix C.6.5). The high gas oil consumption at the beginning of the year is due to the operation of the auxiliary engines supplying some of the load to the switchboard. The marked drop can be seen when the modification to the switchboard allowing the PTO to be placed in parallel, which allows the auxiliary generators to be shut down.

It can be seen that during a 24 hour period the heat loads can fluctuate as shown in figure 6.3 Cylinder water heat load, figure 6.5 Lubricating heat load, figure 6.6 Charge air heat load. The reasons for this lower engine load are attributed to the low trawling speeds, an efficient refrigerating plant heat exchanger system on the refrigerant side, prudent use of engine room machinery and the fact that the fishing grounds are close by and the need to steam long distances in short periods is not required.

Assumption (iv) was based on the specification (Fishing Trawler (b):135) that one 700 kVa alternator (PTO) was supplying plant at 90% load and the second 700 kVa alternator supplied the trawl winch. Although all machinery was operating with the refrigerating plant and fish meal plant at maximum load, the load on the alternators did not exceed 480 kW with the trawl winch off. The trawl winch operation is intermittent with the main load coming on during the hauling of the net. No further reason for the lower electrical load can be found as the monitored load includes the electric fuel heaters.

Assumption (v) was based on an efficient feed system raising the feed water temperature to the auxiliary boiler resulting in a higher output and steam generated by engine exhaust gas only.

The assumption was thus made that there would be sufficient steam generated to drive a small turbine for electrical power. The output of the boiler would be increased due to the raised feed temperature, which would be recovered from the main engine cooling water and exhaust gas. The initial theory assumed the conversion of the sea water cooling system to feed water which would recover heat from the main engine heat exchangers. Due to the system having a turbine exhaust operating at a vacuum, the feed temperature would start at approximately 28°C and pass through the heat exchangers. (Appendix F - DMF 001/89)

The limitations placed on this design are numerous and make the system impractical for the following reasons:

6.2.5.1 For full heat dissipation in the heat exchangers to take place as laid out in Appendix C.6.4 the volume flow must be in the region of 80 m^3/hour per heat exchanger.

6.2.5.2 The plant steam requirements including steam turbine and fish meal plant would be approximately 5400 kg/hour, indicating a large excess of feed water at a high temperature.

6.2.5.3 Mixing after the condenser would not be feasible as the bypassed quantity would be the predominant fluid in the heat balance equation, making the circulating feed water at a higher inlet temperature to the first stage feed heater. To reduce this feed temperature would require a heat exchanger to remove excess heat which would defeat the purpose of the heat recovery system.

The bypassed feed water system could be utilised as a heating source (in other words, for fuel settling tank heating) but the heat value is of such low quality that any heating coil would be very long with a high attendent pressure drop. Such a heating coil would not be cost effective although the coil could be manufactured from mild steel to reduce expense.

Gallois (1981:11) of SEMT Pielstick and results concluded in (Anon 1978:10) stated that for efficient steam recovery, the preheat temperature of the feed water must be in the region of 120°C, with the exhaust from the boiler at 180°C. The exhaust temperature should not be lower than 180°C to prevent acid corrosion of the boiler tubes and uptakes. Using a SEMT Pielstick PC4 diesel engine as his working example with an output power of 7170 kW Gallois (1981:08) claims that it is possible to generate sufficient steam to supply the entire electrical demand, in other words, 4% of the main engine power for a non refrigerated vessel of 7200 kW from exhaust gas heating only.

The power generated is generally 9-10% of the engine output power using a centralised cooling system with an HT and LT circuit. The system will be effective for a steady state engine operation with no fluctuations in engine load. Gallois (1981:09) concludes that heat recovery which enables an exhaust gas boiler to supply the entire electric demand is possible if the propulsion power is above 6000 kW. According to Appendix C.6.4 the output power of the Sulzer main engine fitted to the Roxana Bank has an output of 3180 kW. With the engine running at 60% load it can be seen that the exhaust gas mass flow is reduced considerably which will result in a loss of heat recovery in the exhaust gas boiler. Furthermore, applying the rule of 9% of main engine power for electrical generation, the generated power that could be expected on the Roxana Bank would be 286 kW and that would be at maximum output of 3180 kW.

It is important to note that at this stage the exhaust gas boiler fitted to Gallois' engine receives all the exhaust gas from the main engine exhaust after it has passed through the turbocharger and the steam pressure is 1000 kPa. The high pressure is due to the high inlet feed temperature of 120°C and a differential of 350° - 180°C = 170°C in the exhaust gas temperature as it passes through the boiler and an exhuast gas mass flow of approximately 14,55 kg/second. Similarly, Morton (1981:17) has calculated that a vessel of 2465 kW/cylinder satisfying the conditions as laid out by Gallois can generate approximately 220 kW/cylinder electrical power, recovering heat from the charge air cooler, jacket cooling water and main engine exhaust gas. Thus for a continuous service rating of 20800 kW an output of 2443 kW could be expected after deducting losses. In this case, the mass flow of the exhaust gas is 61,1 kg/second illustrating the effect the gas mass flow rate has on heat transfer in an exhaust gas boiler.

The effect of this exhaust mass flow can be illustrated by using the projected mass flow of the Sulzer 6ZL 40/48 installed on the Roxana Bank. Referring to Appendix C.6.4, the mass flow rate of the exhaust gas is 5,83 kg/second at 85% MCR. The projected boiler output on the exhaust gas heating only is 800 kg/hour.

The mass flow rate is calculated at 85% power when in reality the monitored main engine load indicator fitted on the Roxana Bank was at a maximum of 65%, resulting in a mass flow of:

Volume flow/second $= (\pi/4) * d_c^2 * l_s * (N/60) * Cyl$

$= (\pi/4) * 0{,}4^2 * 0{,}48 * (500/60) * 6$

$= 3{,}018\ m^3/sec$

Converting this to mass flow using the universal gas equation

$$PV = mRT$$

$$190 * 3{,}018 = m * 0{,}287\ \&* (273 + 40)$$

$$m = 6{,}38\ kg/sec$$

This is referred to atmospheric conditions of 101,3 kPa, 15°C

$$m = 6{,}38 * (101{,}3/190) * (313/288)$$

$$m = 3{,}69\ kg/sec$$

where d_c = cylinder diameter in metres

l_s = length of stroke in metres

N = rev/min

P = boost pressure in kPa (abs)

V = volume flow in m^3/sec

m = mass flow in kg/sec

R = universal gas constant in kJ/kgK

$$T = \text{air temperature in } ^\circ K$$

The reduced mass flow will result in a lower boiler output which would result in an available heat value - considering the exhaust gas inlet and outlet temperatures as monitored at 60% load on the 09/10 November 1989:

$$Q = M_{ex}C_{pex}\Delta_{tex}$$

$$= 3{,}69 * 1{,}016 * (350 - 250)$$

$$Q_{ex} = \underline{375\ kW}$$

Operating the engine at the projected output for 5,83 kg/second exhaust gas mass flow and having a temperature differential of (350 - 180) = 170°C will give a heat load of:

$$Q_{ex} = M_{ex}C_{pex}\Delta_{tex}$$

$$= 5{,}83 * 1{,}016 * (350 - 180)$$

$$Q_{ex} = \underline{1007\ kW}$$

The increased mass flow in the case of the Roxana Bank will give an increased boiler output with heating from the main engine exhaust gas. Increasing the temperature differential in this case would entail enlarging the exhaust gas boiler, in other words, fitting a newly manufactured larger boiler. However to benefit from the extra 70°C available would result in a substantial increase in heat transfer area as the pinch point becomes smaller.

It has been proved during the theoretical investigation into Waste Heat Recovery (Anon 1978b:69) that the overall heat transfer coefficient U_c is predominantly dependant on the exhaust gas side. The effectiveness can be found from the relationship:

$$E = 1 - \exp\{-C[1 - \exp[-(NTU)(C)]]\}$$

where $C = mC_p$ for the maximum and minimum fluids

and $NTU = (t_i - t_o)/LMTD$ -- (3-27)

The LMTD can be found using the inlet and saturated temperature of the feed water and the difference in exhaust gas temperature.

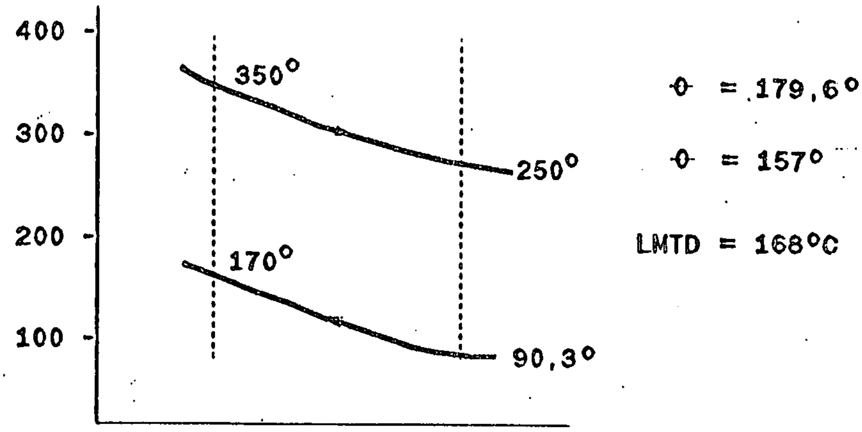


Figure 6.7 Exhaust Gas Boiler Heat Balance

As the exhaust gas is the predominant factor in the heat transfer, the NTU is found by:

$$NTU = (350 - 250)/168$$

$$= 0{,}595$$

Referring to figure 6.9 e% vs NTU Minimum Fluid Mixed the effectiveness of the boiler can be found if the C_{min} is mixed:

where	C_{max}	=	MC_p(feed water)	= 0,92 * 4.,1778	=	3,829
and	C_{min}	=	MC_p(exhaust gas)	= 3,69 * 1.016	=	3,74

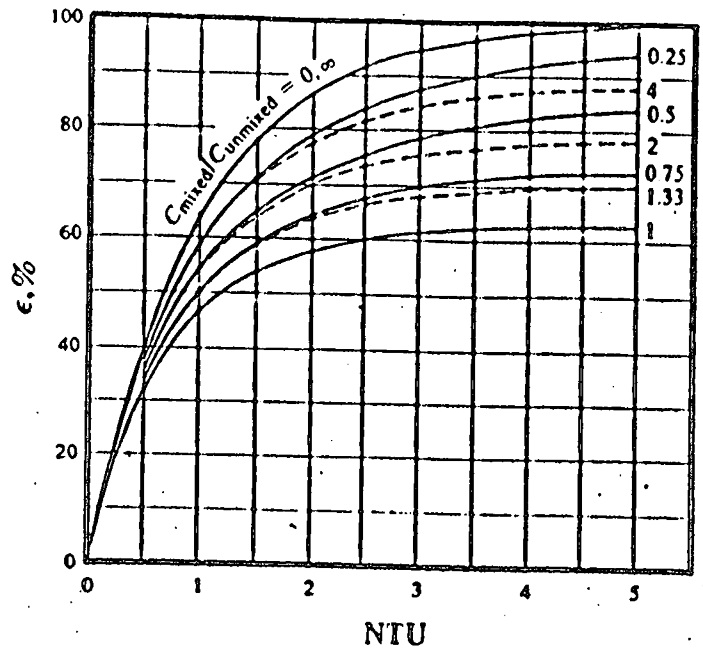


Figure 6.8 E% vs NTU Minimum Fluid Mixed

(Source: Pitts and Sissom, 1977)

The Theoretical Investigation into Waste Heat Recovery (Anon 1978b:69) confirms the dramatic increase in heat exchanger surface area when the pinch point is reduced, in return for a small increase of output, if full use is to be made of the heat available in the exhaust gas. It is due to these reasons that the exhaust gas boiler fitted in the Roxana Bank will be retained.

With the electrical load arrangement on the Roxana Bank being handled by the two 700 kVa PTO's there would be little use for a steam alternator if one was fitted. Again, applying the 9% rule, the theoretical output of a turbo alternator fitted to the Roxana Bank would be:

$$9\% * 85\% * 3180 \quad = \quad 256{,}8 \text{ kW}$$

which if it was available would be insufficient for the whole electrical load and would leave no steam for the fish meal plant, a big user of steam.

The turbine operation discussed assumes that all steam generated is from the exhaust gas only, which is possible with high mass flows of exhaust gas. Table 6.4 Comparative exhaust gas mass flows shows the varying mass flows for different models of Sulzer engines ranging in output power from 3180 to 19400 kW.

Table 6.4 Comparative Exhaust Gas Mass Flows

	Eng Type	rev/min	mass flow	Ouput kW
MFV Roxana Bank	6ZL 40/48	500 (1)	6,27	3180
MV Border	2*12ZV 40/48	500 (2)	2 * 12,56	2 * 6396
SA Winterberg	2*RND90M	122 (3)	2*49,9 (5)	2 * 19500
	2*RND90M	115 (4)	2*29,9 (6)	2 * 13800

Remarks:

(1) Sulzer bulletin states 530 rev/min

(2) Sulzer bulletin states 530 rev/min

(3) Operating conditions before economical steaming

(4) Existing operating conditions with one turbocharger removed - economy steaming.

(5) & (6) The values are calculated from the bore: 900 mm, stroke: 1600 mm and charge air temperature of 45°C. It does not take into account the fuel burnt.

Referring to table 6.4 Comparative exhaust gas mass flows, it can be seen that exhaust gas mass flow has a marked effect on the heat transfer in an exhaust gas boiler application as the exhaust gas temperatures remain similar (Appendix C.6.1). Due to the higher exhaust gas operating temperature of a medium speed diesel engine the exhaust temperature entry into the turbocharger is higher, but the temperatures after the turbocharger are similar. A further increase in exhaust gas temperature after the turbocharger may be obtained by using non watercooled turbocharger casings. Gallois (1981:11) claims an increase of 25°C is possible.

The sources of energy losses have been identified and evaluated as to the suitability of heat recovery sources. The specifications for the heat exchangers are shown in Table 4.3 Heat exchanger specifications and for steady state conditions where a feed water inlet temperature of 73°C will assist in reducing fuel consumption, albeit the quantity saved is small. Further heat recovery is possible from the excess feed water, but the limitations placed on this system such as a shut down evaporator, and low feed temperature from the HTHRU will negate any performance increases that will occur.

Due to engine fluctuations and the limitations placed on the heat recovery system proposed and designed for the Roxana Bank a fuel saving estimate has not been carried out. Furthermore, accurate fuel consumption figures over a short period of time are impossible due to the lack of fuel flow meters in the fuel systems of the Roxana Bank. Also during the short period available when on board during actual operating procedures, boiler burner modulating times were not taken, but in the light of the difficulties that will be experienced in obtaining a steady state heat recovery system, they are now no longer necessary.

Assuming steady engine operating conditions on the Roxana Bank, the following system will be operable:

The four heat exchangers represent the heat recovered from the jacket cooling water and the excess feed water. Exchanger # 1 is a THE from Transheat and exchangers #2, #3 and #4 are PHE's supplied by Alfa Laval.

Referring to heat exchanger #4:

The idea is to economise on the use of steam to reduce the boiler's reliance on the oil fired section and a source of economy is the #2 FW distiller albeit a small saving. It is envisaged that the sea water feed into the distiller be heated by the excess boiler feed water. The ejector water will pass through this heat exchanger as the feed water to the distiller is bled from this supply. With the shell under vacuum in other words - -95 kPa, the sea water will flash off generating water. It must be noted at this stage that the majority of feed water heated will be ejected overboard via the ejector system which will reduce the overall plant efficiency when balancing heat gained and heat rejected. An alternative to this would be to allow only the feed water to pass through the heat exchanger which would reduce the size and the cost of the heat exchanger considerably as the heat exchanger is made from titanium. A problem with this system is that as the distiller produces a maximum of 12 tonne/24 hours, the feed water flow would be in the order of 0,139 kg/second which would result in low Reynolds numbers in the heat exchanger. This problem can be overcome by increasing pressure drop which would have the effect of increasing the fluid velocity. However, increasing pressure drop in the low flow side to acceptably high levels to promote heat transfer will produce unacceptably high pressure drops in the high flow side. However, with a temperature rise of 40°C the heat gained by the sea water would be minimal when considering a fluid flow of 0,139 kg/second or even double this figure and using the heat balance formula:

$$Q_{sw} = M_{sw} C_{psw} \Delta_{tsw}$$

the heat load will be reduced and again from the heat balance, the temperature reduction in the feed water side will not drop sufficiently for a suitable heat balance to take place about the hotwell.

To operate with acceptable heat recovery, the feed water pumped from the hotwell must be of a sufficiently low temperature before entering the HTHRU. Referring to the original subproblem two, the low temperature reached would be acceptable had steam turbine been viable, although this fluid would come from the condenser.

Feeding a temperature of 29,3C would result in a low heat recovery outlet temperature from the HTHRU. In this case, the hotwell temperature - conducting a heat balance about the hotwell, and assuming the steam condensate returning from the fish meal plant, lubricating oil and fuel heaters - will give an outlet temperature to the HTHRU of 35,4°C. The heat balance diagram is shown in figure 6.10 Heat balance about hotwell and has taken into account 10 m^3/24 hours of cold water make up at 21°C.

thus $(2{,}69 * h_f) = (2{,}00 * 29{,}3) + (0{,}12 * 21) + (0{,}57 * 60)$

$h_f = \underline{35{,}4°C}$

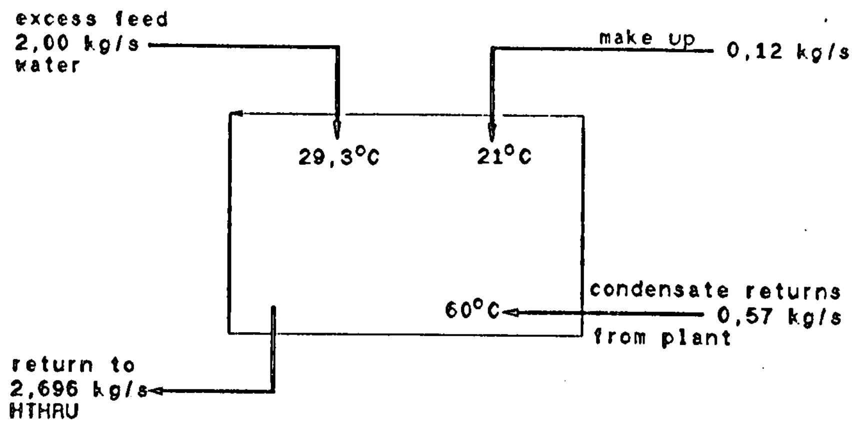


Figure 6.9 Heat Balance About the Hotwell

The temperature required into the HTHRU must be 40°C from the design considerations to ensure adequate heat recovery and an outlet temperature of 73°C, bearing in mind that any higher outlet temperature from the HTHRU will require a greater heat transfer area and will starve the FW distiller.

To obtain an outlet temperature of 40°C, the excess feed water temperature must be adjusted to suit the outlet.

The inlet temperature of the feed must be:

$$(2{,}69 * 40) = (2{,}00 * h_f) + (0{,}12 * 21) + (0{,}57 * 60)$$

$$h_f = 35{,}4°C$$

The temperature of 35,4°C will influence the specifications of the feed water/sea water heat exchanger quite dramatically, reducing the heat transfer surface area and hence the number of pairs of plates, thus reducing the initial cost of this unit.

In the application of exchangers #1, #2 and #3, pressure drop is critical and has been maintained at a minimum. Applying the PHE theory - heat exchangers are large (surface area) to maintain minimum pressure drop. These heat exchanger sizes can be reduced considerably by increasing pressure drop, as pressure drop has a marked effect on heat transfer.

The pressure drop will be maintained at these low values particularly the lubricating oil heat exchanger as the purifier feed pump cannot tolerate high pressure drops and a small relatively inexpensive pump will be used for the settling tank fuel circulating.

The heat transfer characteristics of the excess feed water system designed for the Roxana Bank are full dependent on the temperature of the fluid leaving the HTHRU, which will vary as the engine load fluctuates.

The inlet temperature of the excess feed water is entirely dependent on the continued use of the heat exchanger #1, #2 and #3 at all times. Taking out of service of #2 and #3 will not have a marked effect on the feed temperature as the temperature differential is 1° to 1,5°C. However, should the sea water heater be placed out of service for whatever reason, for example - full water tanks, the fluid temperature entering the hotwell would be in the region of 70°C which would raise the temperature of the hotwell outlet to 65,7°C, reducing the amount of heat recovered to 227 kW assuming a heat exchanger of infinite length and the outlet feed water temperature reached the cylinder cooling water inlet temperature.

The reduced heat recovery would have an effect on the jacket cooling water outlet temperature from the HTHRU with a higher inlet temperature to the FW distiller and with more flow through the central cooler.

Based on steady state operating conditions, a steady heat recovery is possible from the cylinder water and if required from the LT circuit.

With the total boiler output of 3300 kg/hour (0,92 kg/sec) at full design output, the feed water flow into the boiler should not exceed the output leaving a surplus of feed water of 1,79 kg/second. To obtain good heat recovery, the feed water flow has been selected as 10 m^3/hour, thus the feed water flow is 2,71 kg/second. With the excess feed water, the circuit design calls for heat recovery from this excess feed water as shown in Appendix F: DMF006/89.

With the feed water inlet temperature of 40°C to the boiler at present and the operating pressure of 800 kPa (absolute), the quantity of heat required per kilogram to raise the feed water to dry saturated steam is:

reference conditions 800 kPa (absolute)

(Source: Appendix D.1.1)

$$h_g = (721 - 169) + 2047$$
$$= \underline{2599 \text{ kJ/kg}}$$

$$h_g = (721 - 307) + 2047$$
$$= \underline{2451 \text{ kJ/kg}}$$

With a boiler output of 0,71 kg/second (2556 kg/hour), the total heat required for dry saturated steam is 1747,3 kJ/sec. The exhaust gas will supply part of this heat:

Thus $$Q_{eg} = M_{eg} C_{peg} \Delta_{teg}$$
$$= 3{,}69 * 1{,}016 * 100$$
$$Q_{eg} = \underline{374{,}9 \text{ kW}}$$

The remaining 1372,4 kW will therefore be supplied by the oil burner. The heat required from the fuel, assuming a feed inlet temperature of 120°C will be:

reference conditions 800 kPa (absolute)

(Source: Appendix D.1.1)

$$h_g = (721 - 505) + 2047$$
$$= \underline{2263 \text{ kJ/kg}}$$

With the available from the exhaust gas being 323,9 kW, the quantity to be supplied by the fuel will be:

$$\{0{,}71 * 2263\} - 374{,}9 = \underline{1231{,}8 \text{ kW}}$$

which realises a saving of 140,6 kW.

Clearly, from the operating parameters of the fitted steam generation plant fitted to the Roxana Bank and the steam requirements of the fish meal plant, the exhaust gas section does not generate sufficient steam, resulting in the burner modulating. This was confirmed during the monitoring period on board the Roxana Bank in November 1989. Unfortunately, due to time restraints and the short time spent on board the Roxana Bank, burner modulation periods are not available as is the fuel consumed, but during full fish meal plant operation, the boiler fuel consumption is estimated to be in the order of 0,037 kg/second (133,4 kg/hour).

It is thus obvious that for a heat recovery system to function efficiently, the load and operating conditions are methods resulting in lower electrical loads is that for vessels fitted with main engines coupled to PTO's such as electrical alternators, as the electrical load decreases, so does the engine load. An argument to support this would be that there would now be more power available for propulsion - which holds good for some types of vessel, but in a fishing trawler such as the Roxana Bank which has fixed operating parameters and conditions such as speed are not critical, the reduction in power will certainly reduce the fuel consumed, but will also reduce the exhaust gas mass flow causing a drop in heat transfer rate in the exhaust gas boiler.

A vessel operating with a direct drive propeller and a low speed engine not fitted with a PTO will benefit from a lowered electrical load as this will not affect the main propulsion engine. This means that the exhaust gas mass flow rate will not alter and with a reduced electrical load, the exhaust gas boiler would be able to generate sufficient steam to supply steam driven turbo alternator.

Typical examples of full heat recovery systems which recover heat from the charge air coolers, cylinder water and exhaust gas of the main engine, including steam alternators are the contain ship MV Vaal and the VLCC Tokyo Maru.

Table 6.5 Variation in vessels heat loads (pg.164) illustrates the difference in heat exchanger heat loads and hence recoverable heat from main engine heat exchangers based on data from the MFV Roxana Bank, MV border and SA Winterberg.

The research proposal time and cost budget required a monitoring period at sea on the Roxana Bank for three weeks in June, July 1989. At this stage it would have become apparent that fluctuations in engine load would seriously hamper efficient heat recovery on the Roxana Bank. There are several reasons for not attending the Roxana Bank in July/July and only attending the vessel in November 1989. Love (not dated:9) has stated that setting target design objectives should be completed before finding the solutions should any problems occur. This objective must state what is achieved rather than how. With these factors in mind, the difficulties in designing an acceptable feed system were many and varied as discussed. It was thus only in September 1989 that a workable and efficient feed and heat recovery system was designed for the Roxana Bank and October 1989 before the selection of the heat exchangers was completed. The Roxana Bank was attended in May 1989 in Cape Town to ascertain logbook data for the previous months. It was during this period from May 1989 that the owners substituted the South African engineers for Polish engineers with their different methods of operating the machinery in the Roxana Bank. The return voyage from Cape Town in May 1989 was made on the MV Border to obtain data for comparison purposes.

To have an overall comparison with regard to heat loads, mass flows and operating temperatures, the coastal voyage on the SA Winterberg was arranged to obtain data from this vessel and to find a suitable I & J trawler that would be able to rendezvous with the Roxana Bank which was at sea. The bad weather played a major part in the delay in finding the Roxana Bank at sea so that a transfer could be made, and the original proposed period of six days to remain on board was reduced to 30 hours as arrangements had to be made in advance to return to Cape Town on another trawler.

6.5 Variation In Heat Load

		unit	MFV Rox Bank	MV Border	SA Winterberg
JACKETS	Eng out	°C	75	83/84	88/90
	Dist in	°C	73	nil	off
	Dist out	°C	70	nil	off
	HE in	°C	70	83/84	88/90
	HE out	°C	44	no record	41/46
	Eng in	°C	64	78/78	78/81
	Mass flow	kg/s	22,65	45,4	98
	Heat load	kW	593	1000	6450
LUB OIL	Eng out	°C	65	70	45
	HE in	°C	65	70	45
	HE out	°C	53	no record	36
	Eng in	°C	56	56,5	40,5
	Mass flow	kg/s	17,6	25,72	51
	Heat load	kW	318	660	105
CHARGE AIR	Air in	°C	125	155	105
	Air out	°C	41	45	40
	Press	kPa	95	145	72,5
	SW in	°C	18	no record	32
	Mass flow	kg/s	24,91	47,25	29,9
	Heat load	kW	694	1620	2126
EXH	Eng out	°C	522	580	368
	T/C out	°C	350	390	312
	EGB out	°C	270	no record	275
	Heat load		300	---	1063

Furthermore, as fishing is of prime importance, proceeding to the Roxana Bank's position could only take place at night after trawling was completed and the transfer had to be fitted into the fishing program. The early transfer back to the Begonia which was returning to Cape Town was necessary as the Roxana Bank was moving off to fish in Port Elizabeth waters where no transfer could take place.

Referring the alternative design of a heat recovery system incorporating heat recovery from the jacket cooling water with a bypass flow through the heat exchanger and further heat recovery from the high temperature section of the charge air cooler, heat recovery is possible from the #1 heat recovery unit.

This unit recovers heat from the cylinder cooling water which has a split flow which will give a temperature increase to 80°C to the feed water.

Referring to Appendix A.4.16

Further heat recovery was assumed possible from the high temperature section of the charge air cooler. The required temperature rise is 35°C to bring the feed water inlet temperature to the boiler to 115°C.

Assuming that the air inlet to the charge air cooler is 125°C and with a pinch point of 10°C the calculation for a heat exchanger has been carried out.

The charge air cooler has 19 mm ID tubes place 6 in a row. With a mass flow of 0,71 kg/second the fluid velocity is 0,42 m/second which results in a Reynolds number just out of the transition region which will give a poor heat transfer coefficient. This is reflected in the calculation giving an unacceptably high heat transfer area. Mr G Putnin, Chief Engineer on the MV Vaal has reported that the temperature rise of the boiler feed water after passing through the high temperature section of the charge air cooler is 13°C. (Putnin 1989)

It follows that the system is not practical and even with a temperature rise the heat transfer surface area is too large. A PHE cannot be used in this case as they are not suitable for low air pressures where high pressure drop will restrict air flow.

It must be noted that the air will be in crossflow and the heat transfer coefficient from the air will be the controlling coefficient.

6.3 Conclusion

A number of factors have become apparent during the investigation into the energy losses and methods of heat recovery on the Roxana Bank. Although any system is reliant on steady load conditions, small deviations can be tolerated, but load changes from 60% to 20% do not enhance heat recovery prospects.

Furthermore, assuming steady load conditions, there is a lack of high temperature heat sources to raise the feed water temperature to 120°C, although had the system been practical, heat could have been recovered in an economiser in the exhaust trunk after the exhaust gas boiler.

The raising of the feed water to 120°C would reduce the fuel used by the boiler giving a saving in fuel. Raising the cylinder water to 120°C would increase the feed water temperature and by fitting a single heat recovery unit in the jacket circuit with a bypassed flow would increase the feed water to 110°C. There will be resistance to this temperature increase by owners as it exceeds present manufacturer's specifications and difficulties could result with the distillers and the sizing of the cylinder water cooler and the circulating pump.

Reducing mass flows of the cooling water cannot be considered as fluid starvation could result in thermal overloading and damage.

The energy losses on the Roxana Bank have been identified and heat exchanger specifications have been drawn up. Part of these energy losses can be harnessed provided engine loading is steady. The boiler efficiency will be seriously affected by the fluctuations in the proposed plant for the Roxana Bank.

It is therefore apparent that a heat recovery system on the Roxana Bank is not practical due to frequent engine load changes. However, during the design process, the heat recovery system requires the installation of a high and low temperature system with a central cooler for the main cooling systems on the Roxana Bank.

The engine manufacturers, Wartsila, have improved this system by connecting the HT circuit to the high temperature section of the charge air cooler. This system will only operate efficiently under a steady load, at over 50%.

The HT and LT circuits are an important result of this study and should be installed in spite of the fact that a heat recovery system will not be fitted.

The advantages of fresh water cooling are:

6.3.1 Fouling resistances are lower resulting in an improved heat transfer coefficient.

6.3.2 Less cleaning maintenance and reduced downtime of the equipment, resulting in reduced labour costs.

6.3.3 The risk of corrosion and erosion from harsh elements is drastically reduced, resulting in extended heat exchanger life. The piping, valves and other fittings can be of mild steel in the fresh water system which will reduce the initial capital cost.

The HT and LT circuits combined with the central cooler and variable speed sea water pumps will require a high capital cost, particularly that the central cooler is manufactured from titanium, which is very expensive. A typical example of the cost of a titanium PHE with 39 plates as specified in Appendix A.4.12, 12a, 12b and 12c is in the region of R86000, this is complete with end plates and nozzles.

The advantage is that the central cooler can be placed close to the hull with short runs of pipe and the rest of the cooling systems are fresh water.

The system has been designed by the engine manufacturers with the ideal temperatures in mind with the circulating water at 32°C. The specifications for the central cooler are shown in Appendix A.4.12(a-c).

The cooling systems on the Roxana Bank are ideally suited for a central cooling system and the present operating temperatures need not be altered. The specifications for the Roxana Bank central cooling system are designed on the sea temperature of 22°C as the Roxana Bank operates in these temperatures at all times.

Should the decision be made at a later date to operate in higher sea temperatures, the heat exchanger can be expanded.

The assumption was made that the engine load on the Roxana Bank would always be in the region of 80% when in reality the engine load did not rise above 60%. This has a direct effect on efficient heat recovery systems for the Roxana Bank and coupled with the heat load fluctuations make efficient heat recovery impractical.

It has also become apparent that even though a heat recovery system cannot be fitted, a central cooler should be installed in the Roxana Bank. Table 6.6 Cooling water system modification shows the main equipment required for the conversion to a HT and LT cooling fresh water system.

Table 6.6 Cooling Water System Modification

Existing system

pump load - 1, 2, 3

SW Circulating pump	1
Lubricating Oil pump	2
Jacket Cooling Water pump	3
Lubricating Oil HE	
Jacket Cooling Water HE	
Chairge Air HE	

System with Central Cooler

pump load - 1, 2, 3, 4

Lubricating Oil pump	1
Jacket Cooling Water pump	2
LT Circulating pump	3
SW Circulating pump	4
Lubricating Oil HE	
Charge Air HE	
Central Cooler	

It is obvious that Irvin and Johnson have carried out significant modifications to ensure that the Roxana Bank is operated as efficiently as possible with the existing plant fitted. Any further modification must produce sufficient cost effective justification before any conversion can take place.

Due to the nature of the fluctuations of the machinery plant on the Roxana Bank from interrupted fishing and factor/fishmeal operations, no conversions for heat recovery as laid out in this dissertation can be considered practical at this stage.

For this reason, no costing will be carried out on conversions. The conversion to a HT and LT circuit would in reality not save fuel as the pump load will be similar to the existing circuit found on the Roxana Bank.

Furthermore, the age of the vessel must be considered when evaluating a HT and LT circuit. The Roxana bank is already twelve years old and vessels reaching twenty years old become subject to stringent survey requirements by classification societies and conditions laid down by said societies have a direct bearing on the future of the vessel.

Thus with eight years to go to reach the twenty year special survey and the life of the vessel after this period being uncertain, Irvin and Johnson must look at any major conversion into HT and LT cooling very critically.

6.4 Summary

6.4.1 The assumptions made with regard to steady operating conditions and engine load of 80% are not fulfilled and fluctuations occur which make efficient heat recovery impossible.

6.4.2 Efficient and cost effective heat recovery is directly dependent on fluid and gas mass flows, large temperature differentials and steady state conditions.

6.4.3 The exhaust gas boiler on the Roxana Bank does not utilise the full heat load available from the exhaust gas as the outlet is approximately 250°C, but to increase the heat recovery by dropping the exhaust gas to 1790° would result in an uneconomic large exhaust gas boiler.

6.4.4 Care must be taken when reducing electrical load on an installation that is fitted with PTO's as reduced electrical load reduces the engine load, which in turn reduces the exhaust gas mass flow. This does not affect vessels that have main engines for propulsion purposes only.

6.4.5 The heat recovery on larger vessels takes place with resultant steam generation from

6.4.5 The heat recovery on larger vessels takes place with resultant steam generation from the heat transfer process using exhaust gas from the main engine only. For efficient heat recovery, the boiler feed water must be at least 120°C.

6.4.6 The heat recovered by the FW distiller on the Roxana Bank is already sufficient and Irvin and Johnson are operating the plant as efficiently as possible. Due to the nature of the electrical load and supply, the vessel does not require a further source of electrical power.

6.4.7 Literature search into heat recovery systems was limited as heat recovery in fishing vessels has been limited. As the vessels are becoming more sophisticated, further investigations will be made into heat recovery systems for all types of large fishing vessels. Some heat recovery systems found today are suitable for colder conditions than found off the coast of South Africa. The use of FW distillers is becoming more widespread as trawlers become larger and spend more time at sea.

6.4.8 It is obvious that delays in onboard inspections obtaining data and relying on assumptions caused unnecessary work, calculations and formulations as the onboard inspections should have taken place in June/July instead of November 1989. Earlier inspections would have revealed the fluctuations, but due to other considerations such as comparative voyages on the SA Winterberg and MV Border being delayed and the design of the plant taking longer due to the problem with the excess feed water high temperature, caused the delay.

6.4.9 From the discussion, the recommendations are due to forego a heat recovery system. However, a conversion to a high and low temperature fresh centrally cooled system is recommended for reasons discussed, although this modification will require careful consideration due to the age of the Roxana Bank.

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Appendices to accompany Dissertation entitled

A study to assess the enrgy savings potential

in the ocean going trawler :Roxana Bank"

by

D M FIDDLER

DURBAN

15 October 1990

List of Appendices

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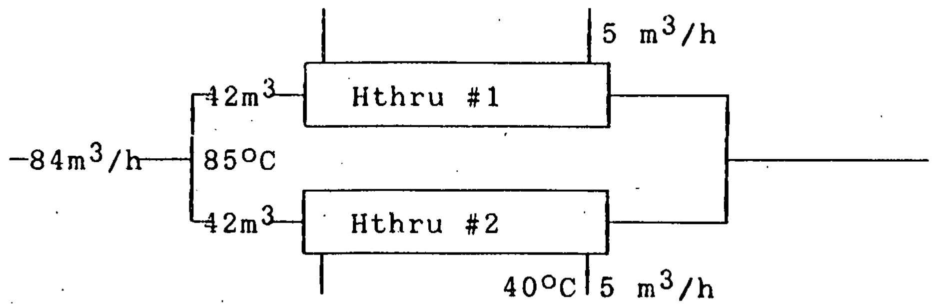
Appendix A:

Appendix A.4.1

High Temperature Heat Recovery Unit



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outer tube: 84 ⎯ 42m^3/h / 42m^3/h reference 85°C (Appendix D.5.4)

density = 971,52 kg/m^3

C_p = 4,198kJ/kgK

Mass flow = 11,334 kg/s Outlet temp = ?

Inner tube: 10 ⎯ 5m^3/h / 5m^3/h reference 57,5°C (Appendix D.1.4)

density = 986,99 kg/m^3

C_p = 4,1775kJ/kgK

Mass flow = 1,371 kg/s Inlet temp = 40,4°C

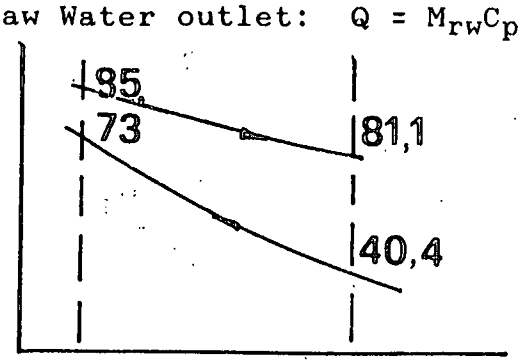
Outlet temp = 73,0°C

Heat balance: Heat recovered/branch $Q = M_{fw}C_{pfw}(T_i - T_o)$ (3-2)

$Q = (5 * 986{,}99 * 4{,}1775 * 32{,}6)/3600$

$= 186{,}7$ kW

Raw Water outlet: $Q = M_{rw}C_{prw}(T_i - T_o)$ (3-2)



$\theta_1 = 12$

$\theta_2 = 40{,}4$

$$LMTD = \frac{12 - 40{,}7}{\ln 12/40{,}7} \quad (3\text{-}14)$$

$= 23{,}49$°C

-- (3-26a)

$$R = \frac{3{,}92}{32{,}6} = \underline{0{,}120} \qquad S = \frac{32{,}6}{85 - 40{,}4} = \underline{0{,}731}$$

-- (3-26b)

R and S are out of range. Use double pipe theory for heat exchanger with multiple inner tubes.

Dimensions (Assume initially that the fluid velocity is 1 m/s)
Nominal pipe size = 4" OD = 114 mm

(Schedule 40) ID = 102 mm

<u>Inner tube</u>: From design configuration n = 37 x 7 mm OD

6,1 mm ID

$$\text{Inner tube flow area} = 37\{(\pi/4 * 0{,}0061^2\})$$

$$A_i = 0{,}00108 \ m^2$$

$$\text{Outer tube flow area} = \{(\pi/4 * 0{,}102^2\} - 37\{(\pi/4) * 0{,}007^2\}$$

$$A_o = 0{,}006747 \ m^2$$

Velocity inner tube = mass flow/tube

= 1,371/37 —— AV = 0,03705

$$0{,}03705 = 986{,}99 * (\pi/4) * 0{,}0061^2 * V$$

$$V = 1{,}284 \ m/s$$

Equivalent diameter (outer tube) = $(4 * 0{,}006747)/\pi * 0{,}102)$

(equation (3-24b) $D_e = 0{,}08422$ m

Fluid velocity (outer tube) = Q = V * A

$$42/3600 = V * (\pi/4) * 0{,}08422^2$$

$$V = 2{,}09 \ m/s$$

Heat Exchanger Design Data

Outer tube:

(1) flow area = 0,006747 m^2

2) $Re = \frac{\rho v d}{\mu}$ ref 82,9°C density = 972,59 kg/m^3

dynamic viscosity = 344 * 10^6 kg/ms

Prandtl No.= 2,15

k = 0,66937 w/m K

(Source:Appendix D.1.4)

$Re = (972,59 * 2,09 * 0,08422)/344 * 10^6$ -- (3-8b)

$= 497645$

3) $Nu = (0,027)Re^{0,8}Pr^{0,333}$ -- (3,13b)

$= (0,027) * 497645^{0,8} * 2,15^{0,333}$

$= 1071,7$

4) $Nu = (h_o d_p)/k$ -- (3-13a)

$= (1071,7 * 0,66937)/0,08422$

$h_o = 8,518\ kW/m^2K$

<u>Outer Tube</u>:

1) Flow area = 0,00108 m^2

2) $Re = \frac{\rho v d}{\mu}$ ref 55,5°C density = 987,28 kg/m^3

dynamic viscosity = 515,93 * 10^6 kg/ms

Prandtl No.= 3,35

k = 0,6447 w/mK

(Source:Appendix D.1.4)

$Re = (987,28 * 1,284 * 0,0061)/51593 * 10^6)$ -- (3-8b)

$= 14988$

3) $Nu = (0,027)Re^{0,8}Pr^{0,333}$ -- (3-13b)

$= (0,027) * 14988^{0,8} * 3,35^{0,8}$

$= 75,39$

4) $Nu = (h_i d_t)/k$ --(3-13a)

h_i = 7,968 kW/m^2K

5) h^{io} = h_i * (id/od)

= 7,968 * 0,0061/0,007

= 6,943 kW/m^2K

Combined Data:

6) Uc = $1/\{(r_o/r_ih_i) + [r_o\ln(r_o/r_i)/k] + 1/h_o\}$ -- (3-5)

= 1/{(0,0035/(0,00305 * 6,943) + [0,0035ln(0,0035/0,00305)/ 0,383] + 1/8,518

= 1/0,2839

Uc = 3,522 kWm^2K

7) $1/U_d = 1/U_c + R_d$ Rd = 0,0015 -- (3-6b)

= 0,2845

U_d = 3,503 kW/m^2K

8) Q = U_dA LMTD -- (3-2)

186,7= 3,503 * A * 23,49

A = 2,268 m^2

9) Add 15% for fouling: (Heat Exchanger Guide:9)

A = 2,268 * 1,15

= 2,608 m^2

10) A = π d n l (surface area)

= 2,608/(π * 0,0061 * 37)

L = 3,678 m

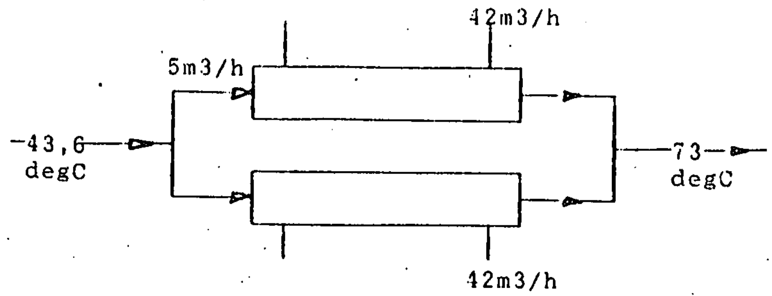
Heat recovery units will consist of 2 units in parallel each having

cylinder water connected in series,

feed water connected in series.

Appendix A.4.2

High Temperature Heat Recovery Unit



		Primary side	Secondary side
Volume Flow	m3/h	2*42 = 84	2*5 = 10
Temperature 1	degC	85	40,4
Temperature 2	degC	81,08	73
Bulk temperature	degC	83,04	56,7
Density	kg/m3	971,52	986,22
Specific Heat	kJ/kgK	4,1980	4,1774
Shell Diameter ID	mm	102,0	-
OD	mm	114,0	-
Tube Diameter ID	mm	-	6,1
OD	mm	-	7,0
No of Shells		1 (2), (4)	-
No of Tubes		-	37 (74) (148)
Passes		1	1
Flow Area	m^2	0,006747	0,00108
Fluid Velocity	m/s	2,09	1,284
Equivalent Diameter	mm	0,08422	-
Dynamic Viscosity	kg ms	$344*10^{-6}$	$515,9*10^{-6}$
Reynolds No		497645	14972
Friction Factor		0,013866	0,0269
Prandtl No		2,15	3,35
Nusselt No		1071,7	75,33
Thermal conductivity	w/mK	0,66934	0,6447
Transfer coeff. shell	w/m^2K	7,306	-
tube	w/m^2K	-	7,903
transfer coeff. corrected	w/m^2K		6,956
COMBINED DATA:			
LMTD	degC	23,49	
Copper Tube Thermal conductivity	w/mK	0,383	
Overall Coefficient	w/m^2K	3,522	
Fouling Factor - Primary		0,001	
Secondary		0,0005	
Final Overall Coefficient	w/m^2K	3,503	
Heat Recovered	kW	186,7	
Heat Transfer Surface Area	m^2	1,928	
Add 15% for Fouling	m	2,268	
Heat exchanger length	m	3,199	
Correction Factor (FT)		1	

Appendix A.4.2a

Transon Heat Exchanger Data: Spiral Spacing 80 mm

TUBE PASS [IN] WATER (SEC)

19 Tubes Cu-Nickel

// Flows	6	S Passes	1
Vel [m/s]	3.7226	Re	53769.2503
Pr	6.1835	Nu	357.7487
Foul	.0300	h in [kW/m2'C	14.3473
LAIN [m]	13.9683	LFIN [m]	9.0723
LW [m]	5.5130		
RLIN [kPa/m]	15.9523	PTIN [kPa]	9.4496

SHELL PASS [EX] WATER (PRIM)

// FLOWS 6 Hs [mm] 80.0

Vel [m/s]	2.2920	Re	220840.1515
Pr	4.5516	Nu	1011.5157
Foul	0.0000	h ex [kW/m2'C	12.6925
Flmtd	1.0000		
LAEX [m]	13.6873	LFEX [m]	0.0000
RLEX [kPa/m]	8.7265	PTEX [kPa/m]	1.8379
LTOT [m]	42.2409		

LMOD [m]	7.0402
MODULE LENGTH	3.5000
No. Series MODS	2.0000
% of LTOT	99.4297

DpIN kPa	130.5654	DpEX kPa	64.7610

DESIGN

6 X 2 US- 19 -3500-Hs80

Appendix A.4.2b

Transon Heat Exchanger Data: Spiral Spacing: 60 mm

TUBE PASS [IN] WATER (SEC)

19 Tubes Cu-Nickel

// Flows	6	S Passes	1
Vel [m/s]	3.7226	Re	53769.2503
Pr	6.1835	Nu	357.7487
Foul	.0300	h in [kW/m2'C	14.3473
LAIN [m]	13.9683	LFIN [m]	9.0723
LW [m]	5.5130		
RLIN [kPa/m]	15.9523	PTIN [kPa]	9.4496

SHELL PASS [EX] WATER (PRIM)

// FLOWS 6 Hs [mm] 60.0

Vel [m/s]	3.3748	Re	267611.2075
Pr	4.5516	Nu	1190.4466
Foul	0.0000	h ex [kW/m2'C	16.0189
Flmtd	1.0000		
LAEX [m]	10.8450	LFEX [m]	0.0000
RLEX [kPa/m]	16.7203	PTEX [kPa/m]	1.8379
LTOT [m]	39.3986		

LMOD [m]	6.5664
MODULE LENGTH	3.5000
No. Series MODS	2.0000
% of LTOT	106.6027

DpIN kPa 130.5654 DpEX kPa 120.7175

DESIGN

6 X 2 US- 19 -3500-Hs60

Appendix: A.4.2c

Transon Heat Exchanger Data: Spiral Spacing: 40 mm

TUBE PASS [IN] WATER (SEC)

19 Tubes Cu-Nickel

// Flows	6	S Passes	1
Vel [m/s]	3.7226	Re	53769.2503
Pr	6.1835	Nu	357.7487
Foul	.0300	h in [kW/m2'C	14.3473
LAIN [m]	13.9683	LFIN [m]	9.0723
LW [m]	5.5130		
RLIN [kPa/m]	15.9523	PTIN [kPa]	9.4496

SHELL PASS [EX] WATER (PRIM)

// FLOWS 6 Hs [mm] 40.0

Vel [m/s]	5.3774	Re	337837.3221
Pr	4.5516	Nu	1451.0940
Foul	0.0000	h ex [kW/m2'C	22.6794
Flmtd	1.0000		
LAEX [m]	7.6601	LFEX [m]	0.0000
RLEX [kPa/m]	37.8962	PTEX [kPa/m]	1.8379
LTOT [m]	36.2137		

LMOD [m]	6.0356
MODULE LENGTH	3.5000
No. Series MODS	2.0000
% of LTOT	115.9783

DpIN kPa 130.5654 DpEX kPa 268.9488

DESIGN

6 X 2 US- 19 -3500-Hs40

Appendix A.4.3

Design of Lubricating Oil Circulating Preheater #1 and #2

Conditions: L.O. circulating 1,1 m^3/h reference 50°C

density = 870 kg/m^3

specific heat capacity = 2,004kJ/kgK

Sulzer recommend 1,1 m3/h circulation (Telecom 19-07-89

R.F.Mueller - Sulzer Bros)

Attempt to raise this temperature to 65°C

$$Q = M_{oil}C_{poil}\,t_{oil} \qquad \text{-- (3-2)}$$

$$= \{1,1 * 870 * 2,004 * (65 - 50)\}/3600$$

$$= 7,99 \text{ kW}$$

Referring to the feed water side: Inlet temperature = 73°C

Mass flow = 1,996 kg/s

reference 73°C (Appendix: D.1.4)

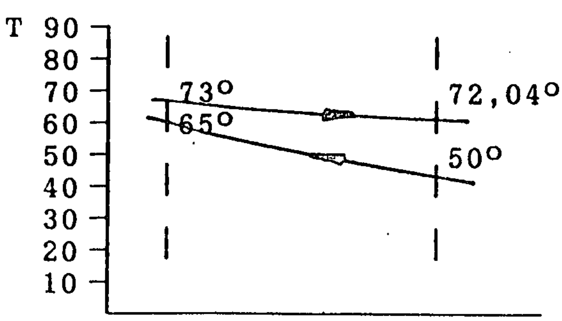
density = 978,05 kg/m^3

specific heat capacity = 4,1894kJ/kgK

$$Q = M_{fw}C_{pfw}\Delta t_{fw}$$

$$7,99 = 1,996 * 4,1894 * (73 - T_o)$$

$$T_o = 72,04°C$$



Hot Fluid		Cold Fluid	
73	high temp	65	8
72,04	cold temp	50	22,04
0,96	difference	15	

R = 0,96/15 = <u>0,064</u>

S = 15/(73 - 50) = <u>0,652</u>

Basing the design of the L.O. heater on double pipe heat exchanger theory as R and S are out of range the fluids should flow:

oil to pass through tubes - feed water to pass through annulus.

Feedwater velocity should be maintained at 1,284 m/s.

1) Shell side reference conditions: 72,52°C

density = 978,31 kg/m^3

specific heat capacity = 4,1981 kJ/kg K

dynamic viscosity = $397{,}8 * 10^{-6}$ kg/ms

(Source: Appendix D.1.4)

2) Tube side reference conditions: 57,5°C

density = 865,54 kg/m^3

specific heat capacity = 2,036 kJ/kg K

dynamic viscosity = $726{,}07 * 10^{-4}$ kg/ms

Velocity of L.O.: $m = \rho Av$

$(1{,}1 * 865{,}54)/3600 = 865{,}54 * A * 1{,}284$

$A = 0{,}000238\ m^2$

$d = 17{,}4$ mm

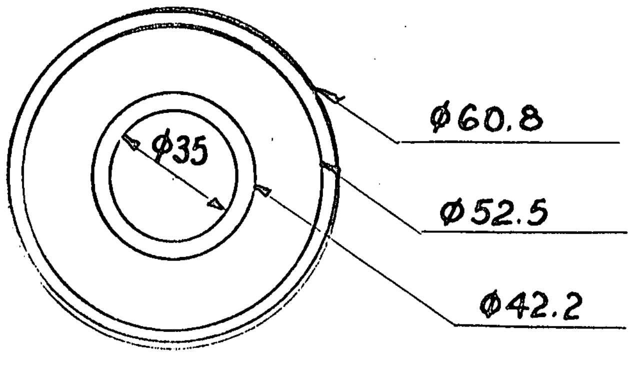
Standard double pipe heat exchanger dimensions: 50,8 x 31,75 mm (2" x 1 1/4")

flow area - pipe = $\pi/4 * 0{,}035^2 = \underline{0{,}00962\ m}$

flow area - annulus $= \pi/4 * (0{,}0525^2 - 0{,}0422^2)$

$= \underline{0{,}000766\ m}$

Sketch showing principle diameters:



The L.O. is to flow in the inner pipe $V_o = 0{,}3176$ m/s

The feed water is to flow in the annulus V_f = 7,36 m/s

Design calculations:

Consider Annulus - feedwater:

1) Equivalent diameter = 4 * flow area)/(π * wetted perimeter)

(Equation (3-24b) = (4 * 0,000766)/(π * 0,0525)

= 0,01858 m

2) Re = $(\rho v d)/\mu$ -- (3-8b)

= $(978{,}31 * 7{,}36 * 0{,}01858)/397{,}8 * 10^{-6}$

= 336306

3) Nu = $(0{,}027)Re^{0{,}8}Pr^{0{,}33}$ Reference 72,52°C

= (0,027) * 336306 * 2,519 Pr = 2,519

= 969,08 k = 0,6612

4) Nu = $(h_o d)/k$ (Source: Appendix D.1.4)

= (969,08 * 0,6612)/0,01858

h^o = 34,486 kE/m^2K

This value is too high - completely out of range. The oil should flow in the annulus and the feed water in the central pipe where the Reynolds number is more acceptable and the feed water side heat transfer coefficient has more realistic values. However, if the L.O. is to flow in the annulus and the feed water in the pipe,the fluid velocity of the L.O. is: V^a = 1,126 m/s and the feed water velocity is V_f = 2,07 m/s.

These values will give a feed water heat transfer coefficient value of h_o = 11,01 kW/m^2 K.

The value of the Reynolds number for the L.O. will be:

Re = $(\rho v d)/\mu$

= $(865{,}54 * 1{,}126 * 0{,}01858)/726{,}06 * 10^{-4}$

$= \underline{249,4}$

From the value of Re the flow is laminar and assuming that the flow of lubricating oil has a fully developed velocity profile, an iterative process must be used to determine the length of an 18,5 mm equivalent diameter tube.

Fluid properties at 60°C are:

density = 865,54 kg/m^3

μ_m = 726,07 * 10^{-4} kg/ms

C_p = 2,036 kJ/kg K

k = 0,1405 w/m K

Pr = 1050

(Source: Appendix D.1.4)

Note: the Prandtl number (Pr) is found from the group $(C_p k)\mu^{1/3}$

With laminar flow the following equations are used to find L:

$$Nu = 3,66 + 0,0668\{(D/L)Re.Pr\}/1 + (0,04)\{(D/L)RePr\}^{1/3}$$

and $Nu = (hd)/k$

from which $h = 14,69 + \{1305/L\}/1 + (0,04)\{4869,4/L\}$

units in w/m K

from $Q = MC_p\Delta T_p = h\pi DL(T_s - T_b)$ where $\pi DL = A$

$h =(1,1 * 865,54 * 2045 * 15)/(3600 * 0,035 * L * \pi * 11,5)$

$= (6412/L)$ w/m K

$(6412/L) = 14,69 + \{1305/L\}/1 + (0,04)\{4869,4/L\}$

Using an iterative process the length can be seen to be in excess of 50 m ({Assuming L = 50 M}

$\underline{\text{ie}\quad 128,24 = 28,77}$

Applying the PHE approximate formula $\theta = (k * 2A)/(m * Cp)$

For LMTD $\theta 1$ = 8^oC

$\theta 2$ = $22,04^oC$

LMTD = (8 - 22,04)/ln 8/22,04 -- (3-14)

= $13,85^oC$

Thermal length θ = Temperature change in primary fluid/LMTD

= (73 - 72,04)/13,85 -- (3-27)

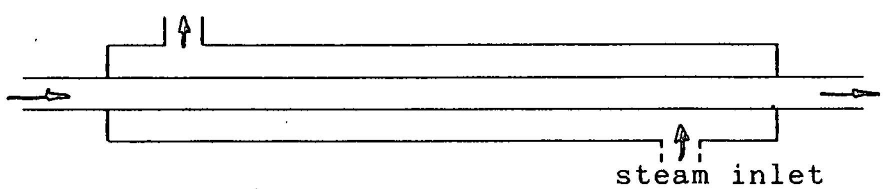
= 0,0693

From the Alsa Engineering Computor Calculation (Appendix A.4.4) the heat transfer surface area is 2,28 m^2 resulting in a PHE Type P2-FM using 21 plates.

Steam Heated Lubricating Oil Preheater

The Lubricating oil enters at 65^oC and exits at 85^oC. Steam enters the heater at 200 kPa (gauge) after being reduced from boiler operating pressure - steam temperature $155,5^oC$

Assuming full counterflow for primary and secondary fluids:



Mass flow of lubricating oil = 0,264 kg/s density = 880,2 kg/m^3

specific heat = 2,003kJ/kgK

Oil Q = $M_{oil}C_{poil}\Delta t_{oil}$ -- (3-2)

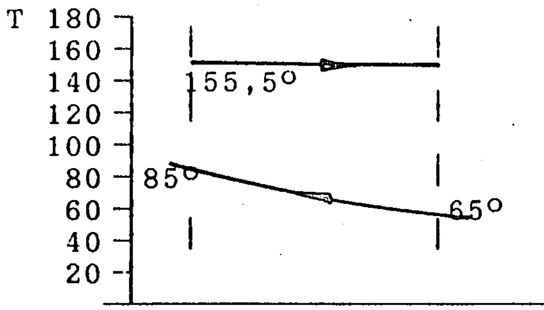
= 0,264 * 2,003 * (85 - 65)

= 10,58 kW

Steam = $(H_g + \{x * H_{fg}\}) * M$

= 0,0043 * (656 + {0,85 * 2103})

= 10,58 kW (Source: Appendix D.1.1)



$\theta 1 = 70,5°C$
$\theta 2 = 90,5°C$

$$LMTD = \frac{(70,5 - 90,5)}{\ln 70,5/90,5} \qquad (3\text{-}14)$$

$$= \underline{80,08}°C$$

Assuming an overall Transfer Coefficient of 113 w/m K (Ud)

$$\text{from } Q = U_d A LMTD$$

$$10,58 = 0,113 * A * 80.0$$

$$A = 1,169 \; m^2$$

1) Lubricating oil in the tubes:

For a double pipe exchanger:

inner tube ID = 0,035 m
OD = 0,0422 m

To find length of double pipe exchanger:

$$A = \pi * D * L$$

$$1,169 = \pi * 0,0422 * L$$

$$L = 8,18 \; m$$

Clearly a multiple single pass heat exchanger is required. Assume U_d = 0,113 kW/m K at the OD of a 7mm tube and using 10 tubes:

$$A = \pi * D * L * N$$

$$1,169 = \pi * 0,007 * L * 10$$

$$L = 5,31 \; m$$

To reduce the heat exchanger to acceptable lengths of 1,5 meters as space is critical, the number of tubes required would be:

$$A = \pi * D * L * N$$

$$1,169 = \pi * 0,007 * 1,5 * N$$

$N = 35,4$ tubes

say 35 tubes of 7 mm od and 6,1 mm id

For fluid velocities in tubes

$m = \ \ Av$

$0,264 = 880,8 * (\ /4) * 35 * 0,0061^2 * v$

$v = 0,2932$ m/sec

Thus $Re = (\ vD)/u$

$= (880,8 * 0,2392 * 0,0061)/726,06 * 10^{-6}$

$= 21,69$

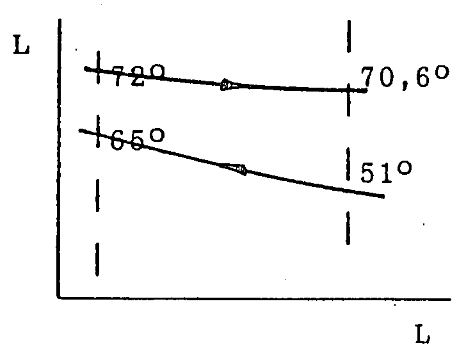
reference 72°C $\rho = 995,0$ kg/m³
$C_p = 4,1888$ kJ/kgK

To find feed water outlet temperature:

$Q_{fw} = M_{fw}C_{pfw}\Delta t_{fw}$

$11,46 = 1,996 * 4,1888 * (72 - T_o)$

$T_o = 70,62°C$



$\theta_1 = 7°$

$\theta_2 = 19,6°$

LMTD = 12,24°C -- (3-14)

Adjust the velocity of the lubricating oil by altering the mass flow of the oil to equal the velocity of the feed water.

Double pipe unit - 50,8 * 31,75 mm (2" * 1 1/4")

Flow area (pipe) $= (\pi/4) * 0,035^2 = 0,000962$ m

Flow area (annulus) $= (\pi/4) * [0,0525^2 - 0,0422^2] = 0,000766$ m

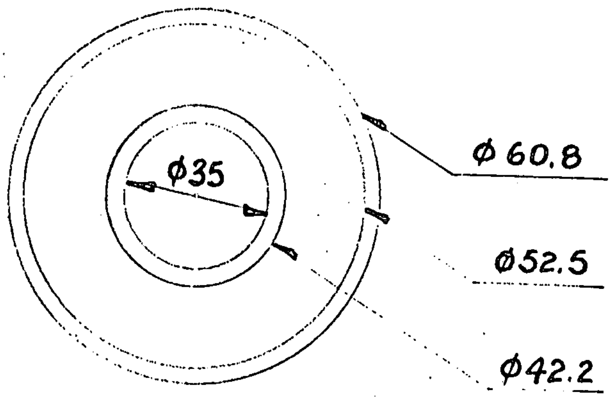
Oil flow in the annulus $Q = V * A$

$= 2,12 * 0,000766 = 5,846$ m³/hour

$$Q = 2{,}12 * 0{,}000766$$

$$= 5{,}846\ m^3/hr$$

Sketch of principle dimensions of pipe and annulus (diameters)



If the fuel circulated for 24 hours heat recovered would be 7,63 kW but as the engine will be running at reduced load there would not be sufficient steam to operate the preheater as well as the fish meal plant which is a 24 hour operation.

Design of double pipe heat exchanger.

Consider pipe (feed water) reference 72°C Pr = 2,54

1) $Re = (\rho v d)/\mu$ -- (3-8b) k = 0,6607 w/mK

$= (978{,}6 * 2{,}12 * * 0{,}035)/400{,}8 * 10^{-6}$

$= 181168$

2) $Nu = (0{,}023)Re^{0{,}8}Pr^{0{,}333}$ -- (3-13b)

$= (0{,}023) * 181168^{0{,}8} * 2{,}54^{0{,}333}$

$= 504{,}82$

3) $Nu = (h_i d)/k$ -- (3-13a)

$= (504{,}82 * 0{,}6607)/0{,}035$

$9{,}529\ kW/m^2K$

Consider pipe - blended fuel:

Diameter = 0,035 m

1) $Re = (\rho VD)/\mu$

ref 70^oC $\quad$ Pr = 1277,5

k = 0,1406 w/mK

= (995 * 0,3186 * 0 035)/726,08 * 10_{-4}

= 152,08

The reynolds number indicates that the flow is streamline (laminar) and again an iterative process wouyld be used. Here a PHE will be fitted to Alsa Engineering specifications after submission of temperature and flow rate programs.

Appendix: A.4.4

PHE specifications: Lubricating Oil Preheater

```
1 P2-FM  AISI 316 0.60mm  21/19pl  A=2.280  Conn 1/1 50mm
10<M=11%  Rf=0.(3.92) Load=7.827 LMTD=13.5  k= 282. 254.
Oil SAE 30 m=0.264 Dp=2.75<75.0 T=50.0=>65.0  1*10 MH
Water      m=2.000 Dp=8.00<75.0 T=73.0=>72.1  1*10 ML

Dp=2.75+0.0030+0.00 v(c)=0.154  P=0.00083 NTU=1.2342 R=69%  S1=>S3
Dp=7.60+0.1538+0.25 v(c)=1.044  P=0.01640 NTU=0.0771 R= 4%  S4=>S2

M=0%  Rf=0.  Load=8.236 LMTD=12.8  k=281.
Oil SAE 30 m=0.264 Dp=2.71 T=50.0=>65.8  1*10 MH
Water      m=2.000 Dp=8.00 T=73.0=>72.0  1*10 ML

Total=2_400 (pl=760) ann.=600

         T(0)  T(NTU/2)    T(1)            .--------1-----------2----
       .-------------------------      m(ch):      0.0264      0.2000
bulk:    50.00    60.37    65.77     v(neck):      0.1066      0.7214
wall:    72.05      291    71.79       v(ch):      0.0486      0.3288
wall:    72.30    10095    71.85  tau(wall):       7.0         19.7
bulk:    73.00    72.35    72.01     t(flow):      11.8         1.7

             Temp    Dens  SpHeat  ThCond    Visc
Oil SAE 30   50.0   881.8   1.950   0.126  52.646
             58.0   876.8   1.978   0.125  36.570
             65.0   872.3   2.003   0.125  27.596
Water        72.0   976.0   4.179   0.663   0.392
             73.0   975.5   4.179   0.664   0.387
             73.0   975.5   4.179   0.664   0.387
```

A = 2,28 m^2		Preheater specification					PHE type:P2 FM		
feed water		P_d 8,00 kPa			lubricating oil P_d 2,71 kPa				
T in	T out	M flow	dens	Sp heat	T in	T out	M flow	dens	Spheat
°C	°C	kg/s	kg/m^3	kJ/kgK	°C	°C	kg/s	kg/m^3	kJ/kgK
73	72	2,00	975,8	4,179	50	65,8	0,264	876,8	1,978

LMTD = 12,8°C   NTU = 1,2342/0,0771
V = 0,154/1,044 m/sec

(Source: Alsa Engineering, Isando, Transvaal)

Appendix: A.4.5

Typical Overall Heat Transfer Coefficients

Fluid	U	
	Btu/hr-ft²-°F	W/m²-K
Oil to oil	30–55	170–312
Organics to organics	10–60	57–340
Steam to:		
Aqueous solutions	100–600	567–3400
Fuel oil, heavy	10–30	57–170
Light	30–60	170–340
Gases	5–50	28–284
Water	175–600	993–3400
Water to:		
Alcohol	50–150	284–850
Brine	100–200	567–1135
Compressed air	10–30	57–170
Condensing alcohol	45–120	255–680
Condensing ammonia	150–250	850–1420
Condensing Freon-12	80–150	454–850
Condensing oil	40–100	227–567
Gasoline	60–90	340–510
Lubricating oil	20–60	113–340
Organic solvents	50–150	284–850
Water	150–300	850–1700

(Source:Pitts D.R. and Sissom L.E. 1977 Heat Transfer Mcgraw Hill Book Company)

Appendix: A.4.6

PHE specifications: Blended Fuel Preheater

```
1 P2-FM  AISI 316 0.60mm  19/17pl  A=2.040 Conn 1/1 50mm
10<M=12%  Rf=0.(3.47) Load=7.827 LMTD=12.5  k= 344. 307.
Oil SAE 40 m=0.264 Dp=14.4<75.0 T=50.0=>65.0  1*5+1*4 L
Water      m=2.000 Dp=40.1<75.0 T=72.1=>71.2  1*5+1*4 L

Dp=14.4+0.0059+0.00 v(c)=0.153  P=0.0043 NTU=1.3455 R=72%  S1=>T1
Dp=39.4+0.3071+0.41 v(c)=1.044  P=0.0821 NTU=0.0840 R= 4%  S4=>T4

M=0%  Rf=0.  Load=8.240 LMTD=11.8  k=342.
Oil SAE 40 m=0.264 Dp=14.2 T=50.0=>65.8  1*5+1*4 L
Water      m=2.000 Dp=40.1 T=72.1=>71.1  1*5+1*4 L

Total=2_800 (pl=680) ann.=730
U-arr=2_600 (pl=900) ann.=630

          T(0)  T(NTU/2)    T(1)            .-------1-----------2----
       .--------------------------         m(ch):     0.0594      0.4500
bulk:     50.00    60.58    65.78       v(neck):     0.2376      1.6224
wall:     71.14      354    70.91         v(ch):     0.1083      0.7395
wall:     71.44    12397    70.97     tau(wall):    18.4        51.1
bulk:     72.10    71.44    71.11       t(flow):    10.4         1.5

            Temp      Dens  SpHeat  ThCond      Visc
Oil SAE 40  50.0     889.6   1.950   0.126    68.849
            58.0     884.8   1.978   0.125    47.013
            65.0     880.2   2.003   0.125    34.997
Water       71.0     976.6   4.178   0.663     0.397
            72.0     976.0   4.179   0.663     0.392
            72.0     976.0   4.179   0.663     0.392
```

A = 2,04 m^2		preheater specifications					Type: P2 FM		
feed water P_d = 40,1 kPa					blended fuel P_d = 14,2 kPa				
T in	T out	M flow	dens	Sp heat	T in	T out	M flow	dens	Sp heat
^oC	^oC	kg/s	kg/m^3	kJ/kgK	^oC	^oC	kg/s	kg/m^3	kJ/kgK
72,1	71,1	2,00	896,3	4,178	50	65,8	0,264	884,6	1,978

LMTD = 11,8^{o}C NTU = 1,3455/0,084
V = 0,153/1,044 m/sec

(Source: Alsa Engineering: Isando, Transvaal)

Appendix: A.4.7

PHE specifications: Distiller Sea Water Preheater

```
1 P2-VB  Titanium 0.60mm  68/66pl  A=7.920  Conn 1/1 50mm
10<M=10%  Rf=0.(0.19) Load=344.1 LMTD=8.9  k= 5394. 4903.
Sea Water m=2.00 Dp=57.9<75.0 T=20.0->63.4  3*11 H
Water     m=2.00 Dp=56.6<75.0 T=30.0<-71.2  1*12+2*11 H

Dp=57.1+0.44+0.40 v(c)=1.00  P=0.11 NTU=5.38 R=86%  S1->T3
Dp=55.7+0.46+0.41 v(c)=1.04  P=0.11 NTU=5.11 R=82%  S4<-T2

M=0%  Rf=0.  Load=349.7 LMTD=8.2  k=5396.
Sea Water m=2.00 Dp=57.9 T=20.0->64.1  3*11 H
Water     m=2.00 Dp=56.6 T=29.3<-71.2  1*12+2*11 H

Frame price n.a.

          T(0)   T(NTU/2)    T(1)              .-------1----------2----
       .----------------------------     m(ch):      0.1818      0.1768
bulk:    20.00     43.51     64.07   v(neck):      0.6297      0.6370
wall:    24.35     12444     67.19     v(ch):      0.2870      0.2903
wall:    25.49     13460     68.30 tau(wall):     49.4         48.3
bulk:    29.33     51.87     71.20   t(flow):      5.9          5.9

            Temp      Dens   SpHeat  ThCond     Visc
Sea Water   20.0    1033.7   3.941   0.590     1.092
            42.0    1025.9   3.967   0.625     0.699
            63.0    1016.6   3.993   0.649     0.501
Water       30.0     994.4   4.182   0.617     0.801
            51.0     986.3   4.173   0.643     0.537
            71.0     976.6   4.178   0.663     0.397
```

Sea water preheater					PHE type: P2 VB				
feed water		P_d = 56,6 kPa			Sea water		Pd = 57,9 kPa		
T in	T out	M flow	dens	Sp heat	T in	T out	M flow	dens	Sp heat
^oC	^oC	kg/s	kg/m^3	kJ/kgK	^oC	^oC	kg/s	kg/m^3	kJ/kgK
71,2	29,3	2,00	986,3	4,173	20	64,1	2,00	1026	3,967

LMTD = 8,2^{o}C   NTU = 5,38/5,11
V = 1,00/1,04 m/sec

(Source: Alsa Engineering: Isando, Transvaal)

Appendix: A.4.8

Calculation of Specific Heat Capacity of Exhaust Gas

Gas	Mole(n)	M	(n)*M	m(nM/Σ nM
CO_2	0,14	44	6,16	0,198
O_2	0,04	32	1,28	0,041
CO	0,02	28	0,56	0,018
SO_2	0,02	64	1,28	0,041
N_2	0,78	28	21,8	0,701

$\Sigma nM = 31,08$

	C_p	dens	$\mu * 10^{-6}$	Pr	k
CO_2	1,0753	0,8938	26,83	0,668	0,043
O_2	1,0037	0,6503	33,92	0,704	0,0478
CO	1,087	0,5685	29,59	0,714	0,0444
SO_2	1,0042	0,5878	29,68	0,696	0,0453
N_2	1,0749	0,5678	29,11	0,686	0,0458

$$C_{pex} = \Sigma mC_p \qquad \text{--(A.4.8a)}$$

$$= (0,198 * 1,0753) + (0,041 * 1,0037) + (0,018 * 1,087) + (0,041 * 1,0042) + (0,701 * 1,0749)$$

$$= 1.0683 \text{ kJ/kgK}$$

$$\rho_{ex} = \Sigma mC_p \qquad \text{--(A.4.8b)}$$

$$= (0,198 * 0,8938) + (0,041 * 0,6503) + (0,018 * 0,5685) + (0,041 * 0\ 5878) + (0,701 * 0,5607)$$

$$= 0,6366 \text{ kg/m}^3$$

$$\mu_{ex} = \Sigma M\mu \qquad \text{--(A.4.8c)}$$

$$= \{(0,198 * 26,33) + (0,041 * 33,92) + (0,018 * 29\ 59) + (0,041 * 29\ 68) + (0,701 * 29,11)\} * 10^{-6}$$

$$= 28,86 * 10^{-6} \text{ kg/ms}$$

$$Pr_{ex} = \Sigma mPr \qquad \text{--(A.4.8d)}$$

$$= (0{,}198 * 0{,}668) + (0{,}041 * 0{,}704) + (0{,}018 * 0{,}724) + (0{,}041 * 0{,}696) + (0{,}701 * 0{,}686)$$

$$= 0{,}6836$$

$$k_{ex} = \Sigma mk \qquad \text{--(A.4.8e)}$$

$$= (0{,}198 * 0{,}043) + (0{,}041 * 0{,}0478) + (0{,}018 * 0{,}0444) + (0{,}041 * 0{,}0453) + (0{,}701 * 0{,}0458)$$

$$= 0{,}0452 \quad w/mK$$

****

Appendix: A.4.9

Design of feed Water Heater (73° - 93°C)

The temperature program: Feed water inlet 73° to 93°C

exhaust condensate inlet 100°C to ?

mass flow feed water: 0,45 kg/s

mass flow condensate: 0,36 kg/s (from fish meal plant at maximum steam consumption)

reference bulk temperature = (73 + 93)/2

reference 100°C = 83°C

C_p = 4,19 kJ/kgK density = 983,79 kg/m^3

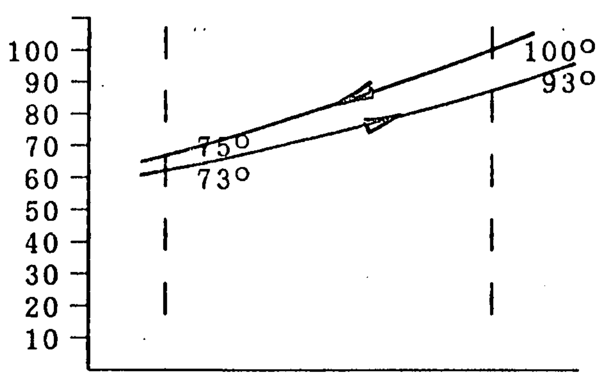
C_p = 4,185 kJ/kgK

(Source Appendix: D.1.4)

$Q = MC_p \Delta t$

$37,66 = 0,36 * 4,19 * (100 - T_o)$

$T_o = 75°C$



LMTD = (2-7)/ln2/7

= 4°C

Assuming a double pipe heat exchanger:

Inner tube ID = 35 mm OD = 42,2 mm

Outer tube ID = 52,5 mm

Flow area - inner tube = $(\pi/4) * d_i^2)$ = $(\pi/4) * 0,035^2$

= 0,000962 m^2

Flow area - outer tube = $(\pi/4)[d_o^2 - d_i^2]$

$= (\pi/4)[0,0525^2 - 0,0422^2]$

$= 0,000766\ m^2$

Velocity of condensate in inner tube: $m = \rho Av$

reference conditions 87,5°C

density = 978,31 kJ/kgK

$\mu = 397,8 * 10^{-6}$ kg/ms

from $m = \rho Av$

$0,36 = 978,31 * 0,000962 * v$

$v = 0,3825$ m/sec

$Re = (\rho vd)/\mu = (978,31 * 0,035 * 0,38250)/397,8 * 10^{-6}$ (3-8b)

$= 32924$ (turbulent flow)

$Nu = (0,027)Re^{0,8}Pr^{0,333} = (0,027) * 32924^{0,8} * 2,519^{0,333}$

$= 151$ (3-13b)

$Nu = (h_i d)/k \quad h_i = (151 * 0,6612)/0,035$ (3-13a)

$= 2,853\ kW/m^2K$

and $h_{io} = h_i * (ID/OD) = 2,366\ kW/m^2K$

Feed water in outer tube:

Equivalent diameter = (4 * flow area)/wetted perimeter

$= (4 * 0,000766)/(\pi * 0,0525)$

$= 0,0186$ m

Again $m = \rho Av$ and $0,45 = 983,79 * 0,000766 * v$

$v = 0,597$ m/sec

$Re = (\rho vd)/\mu = (983,79 * 0,597 * 0,0186)/397 * 10^{-6}$

$= 27468$ (turbulent flow)

$Nu = (0,027)Re^{0,8}Pr^{0,333} = (0,027) * 27468^{0,8} * 2,519^{0,333}$

$= 130,65$

$nu = (h_o d)/k \qquad h_o = (130,63 * 0,6612)/0,0136$

$= 4,643 \text{ kW/m}^2\text{K}$

Using equation 3-5 the Clean Coefficient can be found to be:

$$U_c = 1,687$$

Checking this result against Kerns formula $Uc = (h_{io}h_o)/(h_{io} + h_o)$ gives the value of Uc = 1,567

Adding the fouling factor (15%) $U_d = 1,682 \text{ kW/m}^2\text{K}$

$$Q = U_d A(LMTD)$$

$$37,66 = 1,682 * A * 4$$

$$A = 5,59 \text{ m}^2$$

resulting in a length of: $A = (\ dl)$

$$5,59 = \pi * 0,035 * 1$$

$$L = 50,9 \text{ m}$$

Clearly this length is too large and consideration is given to the fitting of a multitube heat exchanger. However considering equation (3-8) the Reynolds number is in the transitional zone for condensate in a 37 tube heat exchanger and with a 102 mm shell and laminar for the feed flow in the shell. Considering a PHE:

Using the equation (3-27), θ = 5 HTU which gives a heat transfer surface area of 1,048 m^2.

The following PHEs are specified:

Type A3 (maximum flow 9 m^3/hour) with a maximum working temperature of 100°C. This combined with a heating surface area per plate of 0,058 m^2 will give a total number of plates = 18 and velocities of 0,9351 and 0,7497 m/sec.

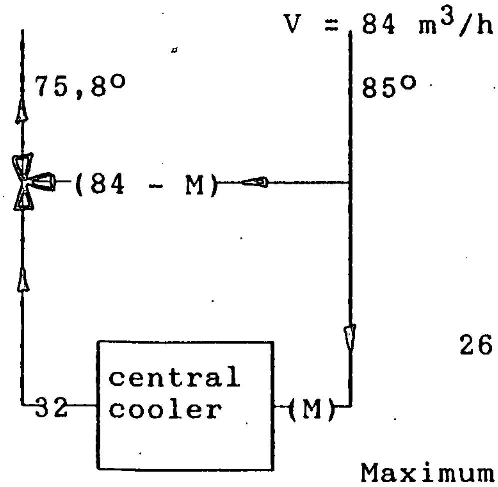
(Source: Heat Excahnger Guide:79)

Appendix: A.4.10

Engine Cooling Water System.

1) Maximum flow through Central cooler assuming no heat recovery.

{heat recovery unit and and fresh water distiller bypassed}



Ref 75,8°C h_f = 318 kJ/kg

85,0°C h_f = 356,1 kJ/kg

32,0°C h_f = 134,2 kJ/kg

(84*318)={(84-M)*356,1}+(M*134,2)

26712 = 29912,4 - 356,1M +134,2M

M = 14,42 m^3/hour

Maximum Flow through Central Cooler

= 14,42 + 80

= 94,42 m^3/hour

Heat balance into Central Cooler: Ref 47,2°C h_f = 197,9 kJ/kg

85,0°C h_f = 356,1 kJ/kg

$(94,42 * h_f) = (80 * 197,9) + (14,42 * 356,1)$

h_f = 222 kJ/kg

Ref.temp = 53,3°C

<u>Heat load Central Cooler</u>: Bulk temp = $\frac{53,3 + 32}{2}$

= <u>42,65°C</u>

Ref 42,65°C density = 993,41 kg/m^3

specific heat capacity = 4.1780 kJ/kgK

$Q = M_{rw}C_{prw}(T_i - T_o)_{rw}$

= (94,42 * 993,41 * 4,1780 * [53,3 - 32])/3600

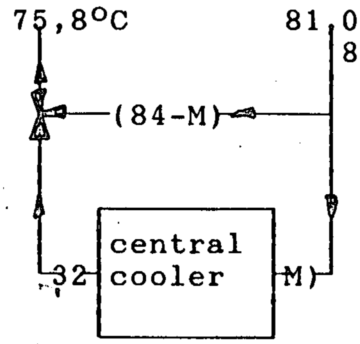
= 2318,6 kW

From ideal conditions - heat to be dissipated = 1410 + 896

= 2306 kW

Engine cooling water system

System operating with HTHRU in use and with FW Distiller bypassed.

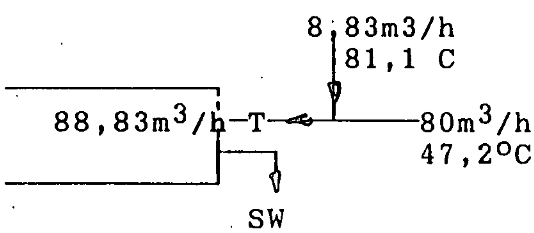


Ref 75,8°C h_f = 318 kJ/kg
81,1°C h_f = 339,6 kJ/kg
32,0°C h_f = 134,2 kJ/kg

(84 * 318) = {(84 - M) * 339,6}+(M*134,2)
26712 = 28526,4 - 339,6M +134,2M

M = 8.83 m^3/h

For Temperature into Central Cooler



Ref 81,1°C h_f = 339,6 kJ/kg
47,2°C h_f = 197,9 kJ/kg

(88,83*hf)=(8,83*339,6)+(80*197,9)
h_f = 211,98 kJ/kg
T = 50,64°C

Heat load Central cooler: Bulk Temp = $\frac{50,64 + 32}{2}$
= 41,32°C

Ref 41,32°C: density = 994,0kg/m^3

Specific heat capacity = 4,1781kJ/kgK

Volume flow = 88,83 m^3/h

$Q = M_{rw}C_{prw}(T_i - T_o)_{rw}$

= {88,83 * 994 * 4,1781 * (50,64 - 32)}/3600

= 1910 kW

Actual heat to be dissipated = 1410 + {896 - (2 * 186,7)}

(Ideal conditions) = 1932,6 kW

Engine cooling water system

Heat recovered in HTHRU and #1 FW Distiller.

1) Heat recovered HTHRU = 2 * 186,7 = 373,42

Heat recovered in #1 FW dist. = 442 = 442,0

815,4

Outlet temperature from HTHRU = 81,1°C

Temperature conditions after #1 FW Distiller

Ref 81,1°C : density = 973,34kg/m^3

Specific heat capacity = 4,1947kH/kgK

$$Q = M_{cw}C_{pcw}(T_i - T_o)_{cw}$$

$$442 = [55 * 973,43 * 4,1947 * (81,1 - T_o)]/3600$$

$$T_o = 74°C$$

Heat balance about (A) Ref 81,1°C hf = 339,6 kJ/kg

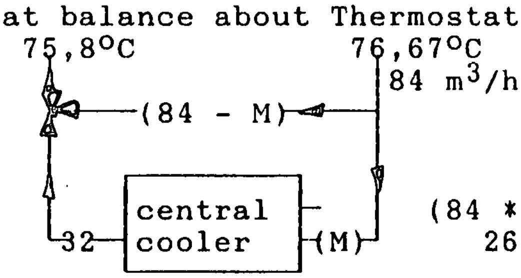
74,0°C h_f = 311,6 kJ/kg

$$(84 * h_f) = (29 * 339,6) + (55 * 311,6)$$

$$h_f = 321,3 \text{ kJ/kg}$$

$$T(A) = 76,67°C$$

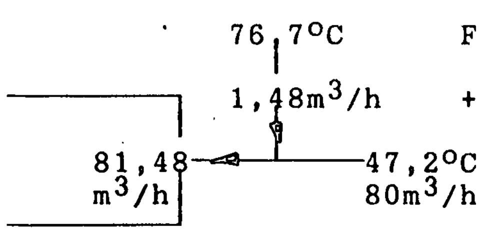
heat balance about Thermostatic valve.



Ref 75,8°C h_f = 318,0 kJ/kg
76,7°C h_f = 321 3 kJ/kg
32,0°C h_f = 134,2 kJ/kg

$$(84 * 318) = \{(84 - M)*321,3\}+(M*134,2)$$
$$26712 = 26989 - 321,3M + 134,2M$$
$$M = 1,48 \text{ m}^3/\text{hour}$$

For temperature into Central Cooler:



Flow through Central cooler

+ 1,48 + 80 = 81,48 m^3/hour

Ref 47,2°C h_f = 197,9 kJ/kg
76,7°C h_f = 321,3 kJ/kg

$$(81,84 * h_f) = (80 * 197,9) + (1,48 * 321,3)$$

$$h_f = 200 \text{ kJ/kg}$$

$$T = 47,7°C$$

Heat load Central Cooler: Bulk temp $= \frac{47,7 + 32}{2}$
$= 39,58°C$

Ref 39,85°C density = 994,65 kg/m^3

Specific heat capacity = 4,1782 kJ/kgK

Vol flow = 81,48 m^3/h

$$Q = M_{rw}C_{prw}(T_i - T_o)_{rw}$$

$$= [81,48 * 994,65 * 4,1782 * (47,7 - 32)]/3600$$

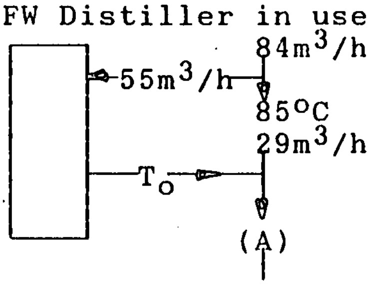
$$= 1476,8 \text{ kW}$$

Actual heat to be dissipated = 1410 + {896 - [(2 * 186,7) + 442]}

(Ideal conditions) = 1490,6 kW

<u>Engine cooling water system</u>

#1 FW Distiller in use with HTHRU bypassed.



Heat recovered by FW Distiller = 442 kW

Ref 85°C density = 970,7 kg/m^3

Specific heat capacity = 4,1986 kJ/kgK

$$Q = M_{cw}C_{pcw}(T_i - T_o)_{cw}$$

$$442 = [55 * 970,7 * 4,1986 * (85 - To)]/3600$$

$$T_o = 77,9°C$$

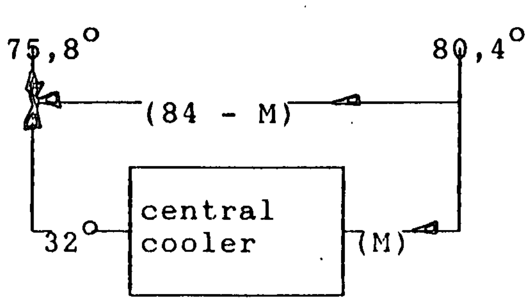
Heat balance at (A): Ref 77,9°C h_f = 326,6 kJ/kg

85,0°C h_f = 356,1 kJ/kg

$$(84 * h_f) = (55 * 326,6) + (29 * 356,1)$$

$$T_o = 336,8 \text{ kJ/kg}$$

$$T(A) = 80,4°C$$



Ref 80,4°C hf = 336,8 kJ/kg

75,8°C hf = 318,0 kJ/kg

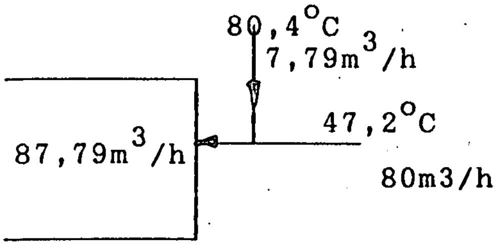
32,0°C hf = 134,2 kJ/kg

$$(84 * 318) = \{(84 - M) * 336,8\} + (M * 143,2)$$

$$26712 = 28291,2 - 336,8M + 134,2M$$

$$M = 7,79\ m^3/hour$$

For temperature into Central Cooler:



flow through Central Cooler:

$$= 7,79 + 80 = 87,79\ m^3/h$$

Ref 80,4°C h_f = 336,8 kJ/kg
47,2°C h_f = 197,7 kJ/kg

$$(87,79 * h_f) = (7,79 * 336,8) + (80 * 197,9)$$

$$h_f = 210,2\ kJ/kg$$

$$T_{in} = 50,23^\circ C$$

Heat load Central Cooler: Bulk temp $= \frac{50,23 + 32}{2}$

$$= 41.12^\circ C$$

Ref 41,12°C density = 994,09 kg/m^3
Specific heat capacity = 4,1781 kJ/kgK
Volume flow = 87,779 m^3/h

$$Q = MrwCprw(Ti - To)rw$$

$$= [87,79 * 994,04 * 4,1781 * (50,23 - 32)]/3600$$

$$= 1846,4\ kW$$

Actual heat dissipated = 1410 + (896 - 442)

= 1864 kW

When not using the heat recovery unit there is heat available to operate #2 FW Distiller in addition to #1 FW Distiller as shown below.

Ref 80,4°C density = 973,82 kg/m^3

Specific heat capacity = 4,194 kJ/kgK

$$Q = MrwCprw(Ti - To)rw$$

$$442 = [55 * 973{,}82 * 4{,}194 * (80{,}4 - T_o)]/3600$$

$$T_o = 73{,}3^{\circ}C$$

Heat balance at (A): Ref $73{,}3^{\circ}C$ $h_f = 307{,}0$ kJ/kg

$80{,}4^{\circ}C$ $h_f = 336{,}8$ kJ/kg

$$(84 * h_f) = (55 * 307) + (29 * 336{,}8)$$

$$h_f = 317{,}3 \text{ kJ/kg}$$

$$T(A) = 75{,}7^{\circ}C$$

This temperature T(A) is lower than the required engine inlet temperature and undercooling of the engine would result.

The Sulzer recommended differential of 10 C is not yet reached, the inlet being 75 C, but with the inherent inability of thermostatic control valves to be accurate to within 2 C plus or minus, severe undercooling would result with the attendent drop in cylinder cooling water outlet temperature resulting in a poor operation of the distillers and a temperature fluctuation that could cause thermal stressing of the engine.

During engine operation, generally the boiler will operate and therefore the feedwater system would operate resulting in the HTHRU being utilised in series with #1 FW Distiller.

NB All the tebale used in Appendix A.4.10 are taken from Appendix D.1.1, D.1.2, D.1.3, D.1.4.

Appendix: A.4.11

Central Cooler Temperature and Volume Flow Program

Heat Load Central Cooler				No heat recovery	HTHRU @ Dist use	HTHRU use Dist bypass	Dist use HTHRU bypas:
High	Vol	Bypass	M3/h	69,58	82,52	75,17	76,21
	Flow	C.Cool	M3/h	14,42	1,48	8,83	7,79
T		HRU in	degC	85	85	85	85
E	T	HRUout	degC	-	81,08	81,08	85
M	E	Dis in	degC	-	81,08	-	85
P	M	Disout	degC	-	74	-	77,9
	P	Mixed	degC	85	76,76	81,08	80,4
	heat	Feed	kW	Nil	373,4	373,4	Nil
Cir	Reco	Dist	kW	Nil	442,0	Nil	442,0
cuit	very	Tot	kW	Nil	815,4	373,4	442,0
C	Vol	Ex HT	M3/h	14,42	1,48	8,83	7,79
e n		Ex LT	M3/h	80	80	80	80
t r	Flow	Total	M3/h	94,42	81,48	88,83	87,79
a l	T	HT	degC	85	76,67	81,08	80,4
	E	LT	degC	47,2	47,2	47,2	47,2
C	M	CC in	degC	53,3	47,7	50,6	50,23
o o	P	CC out	degC	32	32	32	32
l e	Heat	calc	kW	2318,6	1476,8	1910	1846,4
r	diss	Balan	kW	2306	1490,8	1932,6	1846

Appendix: A.4.12

Specification of Heat Exchanger Type A15 - 8FM

(Sea Temperature - 22°C)

		PRIMARY SIDE HOT SIDE	SECONDARY SIDE COLD SIDE
FLUID	:	WATER	NACL 5.0%
REFERENCE TEMPERATURE	C	41.6	29.3
DENSITY	KG/M3	990	1031
RELATIVE DENSITY	:	0.990	1.03
SPECIFIC HEAT CAPACITY	KJ/(KG*K)	4.18	3.95
THERMAL CONDUCTIVITY	W/(M*K)	0.632	0.606
DYNAMIC VISCOSITY	CP	0.635	0.891
VISCOSITY AT INLET TEMP.	CP	0.517	1.04
VISCOSITY AT OUTLET TEMP.	CP	0.767	0.747
KINEMATIC VISCOSITY	CST	0.641	0.864
MASS FLOW RATE	KG/S	26.2	36.1
FLOW RATE	M3/H	95.7	126
INLET TEMPERATURE	C	53.3	22.0
OUTLET TEMPERATURE	C	32.0	38.3
PERMISSIBLE PRESSURE DROP	KPA	200	200
PRESSURE DROP	KPA	121	197
CONNECTION PRESSURE DROP	KPA	1.03	1.88
PUMPING POWER	KW	3.2	6.9
VELOCITY IN CONNECTION	M/S	1.5	2.0
WALL SHEAR STRESS	PA	110	178
VOLUME	L	34	34
EXTREME WALL TEMPERATURE	C	26.1	44.6
NTU, NOMINAL	:	1.7	1.3
NTU, CLEAN CONDITIONS	:	1.9	1.5
THERMAL EFFICIENCY, NOMINAL	%	68	52
THERMAL EFFICIENCY, CLEAN	%	71	54
HEAT EXCHANGED	KW	2330	
L.M.T.D	C	12.3	
O.H.T.C, CLEAN CONDITIONS	W/(M2*K)	7573	
O.H.T.C, SERVICE	W/(M2*K)	6823	
HEAT TRANSFER AREA	M2	27.8	
FOULING RESISTANCE	M2*K/W	0	
DUTY MARGIN	%	11	
FOULING FACTOR	M2*K/W	0.015*10-3	
MARGIN INCL. FOULING RES.	%	11	
MIN. REQUIRED DUTY MARGIN	%	10	
RELATIVE DIRECTION OF THE FLUIDS	:	COUNTERFLOW	
CHANNEL ARRANGEMENT	:	1*(12MH+7L) / 1*(12ML+7L)	
NUMBER OF PASSES	:	1	
NO. OF CHANNELS PER PASS	:	19	
NUMBER OF PASSES	:	1	1
NO. OF CHANNELS PER PASS	:	19	19
NO. OF PLATES	:	39	
NO. OF EFFECTIVE PLATES	:	37	
EXTENSION CAPACITY	PLATES	55	
TOTAL CAPACITY OF THE FRAME	PLATES	94	
PLATE MATERIAL	:	TITANIUM	
PLATE THICKNESS	MM	0.6	

(Source: Alsa Engineering, Isando, Transvaal)

Appendix: A.4.12a

Specification of Heat Exchanger Type A15 - 8FM

(Sea Temperature - $16^{o}C$)

		PRIMARY SIDE HOT SIDE	SECONDARY SIDE COLD SIDE
FLUID	:	WATER	NACL 5.0%
REFERENCE TEMPERATURE	C	37.4	25.5
DENSITY	KG/M3	992	1032
RELATIVE DENSITY	:	0.992	1.03
SPECIFIC HEAT CAPACITY	KJ/(KG*K)	4.18	3.95
THERMAL CONDUCTIVITY	W/(M*K)	0.627	0.600
DYNAMIC VISCOSITY	CP	0.688	0.965
VISCOSITY AT INLET TEMP.	CP	0.570	1.20
VISCOSITY AT OUTLET TEMP.	CP	0.845	0.790
KINEMATIC VISCOSITY	CST	0.694	0.935
MASS FLOW RATE	KG/S	22.6	24.7
FLOW RATE	M3/H	82.3	85.9
INLET TEMPERATURE	C	47.5	16.0
OUTLET TEMPERATURE	C	27.5	35.4
PERMISSIBLE PRESSURE DROP	KPA		
PRESSURE DROP	KPA	92.3	96.8
CONNECTION PRESSURE DROP	KPA	0.765	0.879
PUMPING POWER	KW	2.1	2.3
VELOCITY IN CONNECTION	M/S	1.3	1.4
WALL SHEAR STRESS	PA	83.9	87.7
VOLUME	L	34	34
EXTREME WALL TEMPERATURE	C	21.2	41.3
NTU, NOMINAL	:	1.7	1.6
NTU, CLEAN CONDITIONS	:	1.9	1.8
THERMAL EFFICIENCY, NOMINAL	%	63	61
THERMAL EFFICIENCY, CLEAN	%	66	64
HEAT EXCHANGED	KW	1887	
L.M.T.D	C	11.8	
O.H.T.C, CLEAN CONDITIONS	W/(M2*K)	6322	
O.H.T.C, SERVICE	W/(M2*K)	5747	
HEAT TRANSFER AREA	M2	27.8	
FOULING RESISTANCE	M2*K/W	0	
DUTY MARGIN	%	10	
FOULING FACTOR	M2*K/W	0.016*10-3	
MARGIN INCL. FOULING RES.	%	10	
MIN. REQUIRED DUTY MARGIN	%	10	
RELATIVE DIRECTION OF THE FLUIDS	:	COUNTERFLOW	
CHANNEL ARRANGEMENT	:	1*(12MH+7L) / 1*(12ML+7L)	
NUMBER OF PASSES	:	1	
NO. OF CHANNELS PER PASS	:	19	
NUMBER OF PASSES	:	1	1
NO. OF CHANNELS PER PASS	:	19	19
NO. OF PLATES	:	39	
NO. OF EFFECTIVE PLATES	:	37	
EXTENSION CAPACITY	PLATES		
TOTAL CAPACITY OF THE FRAME	PLATES		
PLATE MATERIAL	:	TITANIUM	
PLATE THICKNESS	MM	0.6	

(Source: Alsa Engineering, Isando, Transvaal)

Appendix: A.4.12b

Specification of Heat Exchanger Type A15 - 8FM

(Sea Temperature - $30^{o}C$)

			Primary side	Secondary side
Fluid	:		Water	SEA WATER
Density	kg/m3		991	1028
Specific heat capacity	kJ/(kg·K)		4.18	3.96
Thermal conductivity	W/(m·K)		0.630	0.616
Viscosity, inlet	cP		0.517	0.879
outlet	cP		0.767	0.650
Mass flow rate	kg/s		26.2	36.1
Inlet temperature	°C		53.3	30.0
Outlet temperature	°C		32.0	46.3
Permissible pressure drop	kPa		200	200
Total pressure drop	kPa		132	199
Connection pressure drop	kPa		2.00	3.68
Velocity in connection	m/s		1.5	2.0
Pumping power	kW		3.5	7.0
Wall shear stress	Pa		59.4	89.1
Volume	l		122	122
Heat load	kW		2330	
L.M.T.D	°C		3.99	
O.H.T.C, clean conditions	W/(m²·K)		6464	
service	W/(m²·K)		5824	
NTU, nominal	:		5.3	4.1
clean conditions	:		5.9	4.5
Heat transfer area	m²		100	
Fouling resistance	m²·K/W		0	
Duty margin	%		11	
Relative direction of the fluids	:		counterflow	
Channel arrangement	:		2*(7H+27MH) / 2*(7H+27ML)	
No. of plates	:		138	
effective plates	:		134	
partition plates	:		1	
Plate material	:		Titanium	
thickness	mm		0.6	
Connection size	mm		150	150
Nozzle orientation	:		S1->T1	T4->S4

(Source: Alsa Engineering, Isando, Transvaal.)

Appendix: A.4.12c

Specification of Heat Exchanger Type A15 - 8FM

(Sea Temperature - 16^{0}C)

		Primary side.	Secondary side
Fluid	:	Water	SEA WATER
Density	kg/m3	996	1032
Specific heat capacity	kJ/(kg-K)	4.19	3.95
Thermal conductivity	W/(m-K)	0.611	0.598
Viscosity, inlet	cP	0.570	1.20
outlet	cP	1.11	0.650
Mass flow rate	kg/s	22.6	24.7
Inlet temperature	°C	47.5	16.0
Outlet temperature	°C	16.2	46.3
Permissible pressure drop	kPa	200	200
Total pressure drop	kPa	101	98.1
Connection pressure drop	kPa	1.48	1.72
Velocity in connection	m/s	1.3	1.4
Pumping power	kW	2.3	2.4
Wall shear stress	Pa	45.4	44.0
Volume	l	122	122

Heat load	kW		2958	
L.M.T.D	°C		0.559	
O.H.T.C, clean conditions	W/(m²-K)		5220	
service	W/(m²-K)		52201	
NTU, nominal	:	56		54
clean conditions	:	5.6		5.4
Heat transfer area	m²		100	
Fouling resistance	m²-K/W		0	
Duty margin	%		-90	

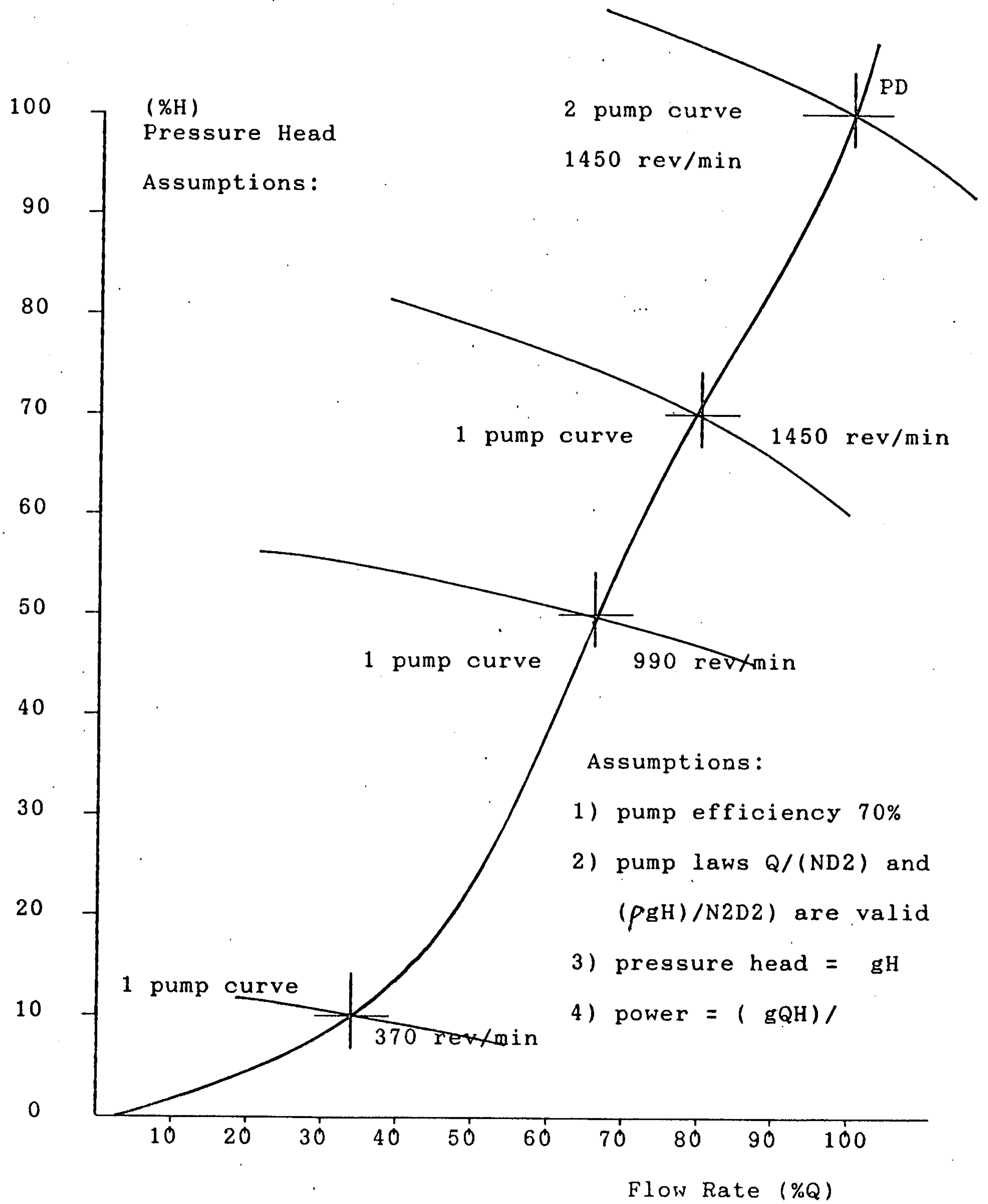
Relative direction of the fluids	:	counterflow
Channel arrangement	:	2*(7H+27MH) / 2*(7H+27ML)
No. of plates	:	138
effective plates	:	134
partition plates	:	1

Plate material	:	Titanium	
thickness	mm	0.6	
Connection size	mm	150	150
Nozzle orientation	:	S1->T1	T4->S4

(Source: Alsa Engineering, Isando Transvaal)

Appendix: A.4.13

Relationship Between Pump Speeds and Volume Flow



Appendix A.4.14

Calculation of Pump Capacities, Rotational Speeds and power requirements

Assumptions: pump efficiency = 70%

pump laws: $Q/(ND^3)$, $(\rho gH)/N^2D^2$, $P = (\rho GQH)/\eta$

sea water density = 1031 kg/m^3

head is constant (H) pump speed is 1450 rev/min

impeller is constant (D)

maximum flow required for maximum raw water flow: 1,35 * 94,42

Vol flow = 130 m^3/h

with a pump speed of 990 rev/min:

$$130/(1450 * D^3) = Q_2/(990 * D^3)$$

$$Q_2 = 88,8\ m^3/h$$

assuming that efficiency remains constant at 70%, and the available head is 20 meters.

Thus H_2: $$(\rho gH_1)N^2D^2 = (\rho gH_2)/D^2D^2$$

$$H_2 = 9,32\ m$$

which gives a pressure of 94,3 kPa

this will give a pumping power required to pump the fluid through the heat exchanger:

$$P = (\rho gQH)/\eta$$
$$= 3,3\ kW$$

By the same method, the static head and pressure head will 1,3 m 13,2 kPa for a pump speed of 370 rev/min. (Appendix A.4.13)

The minimum flow through the central cooler for raw water is 81,5 m^3/h giving a sea water flow of 110 m^3/h.

Using the pump laws will give a pump speed of 1226,9 rev/min.

The minimum condition for the water flow through the PHE with a sea temperature of 16°C as shown in appendix A.4.12c. will give a pump speed of 958 rev/min.

A rotational speed of 958 rev/min gives a static head of 8,73 meters and pressure head of 88,3 kPa using the pump laws.

From Appendix A.4.12c, the minimum head allowed for this volume flow of 85,9 m^3/h = 96,8 kPa which gives a static head of 9,57m.

using the volume flow law:

pump speed $20/1450^2 = 9,57/N^2$

$N^2 = 1003$ rev/min

power = $(\rho g Q H)/\eta$

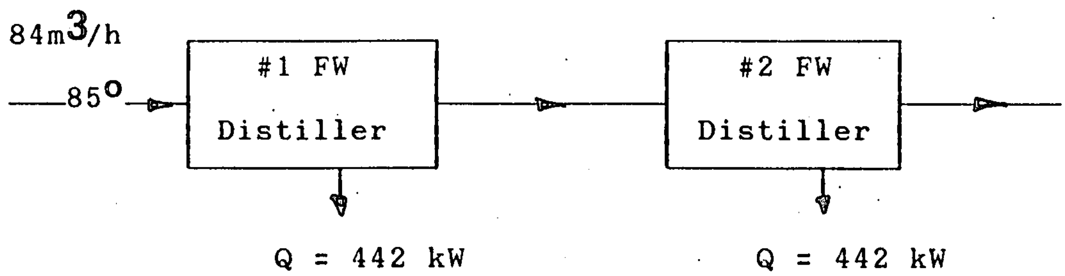
= (1031 * 9,81 * 0,023 * 9,57)/0,7

= 3,3 kW.

****

Appendix: A.4.15

Effect of #1 and #2 FW Distillers in Series



Reference $85^{o}C$ $\rho = 970{,}7\ kg/m^3$

$C_p = 4{,}1986\ kJ/kgK$

(Source: Appendix: D.1.4)

$Q = MC_p\Delta_t$

$442 = \{84 * 970{,}7 * 4{,}1986 * (85 - T_o)\}/3600$

$T_o = 80{,}35^{o}C$

Reference $80{,}4^{o}C$ $\rho = 973{,}86\ kg/m^3$

$C_p = 4{,}1940\ kJ/kgK$

$Q = MC_p\Delta_t$

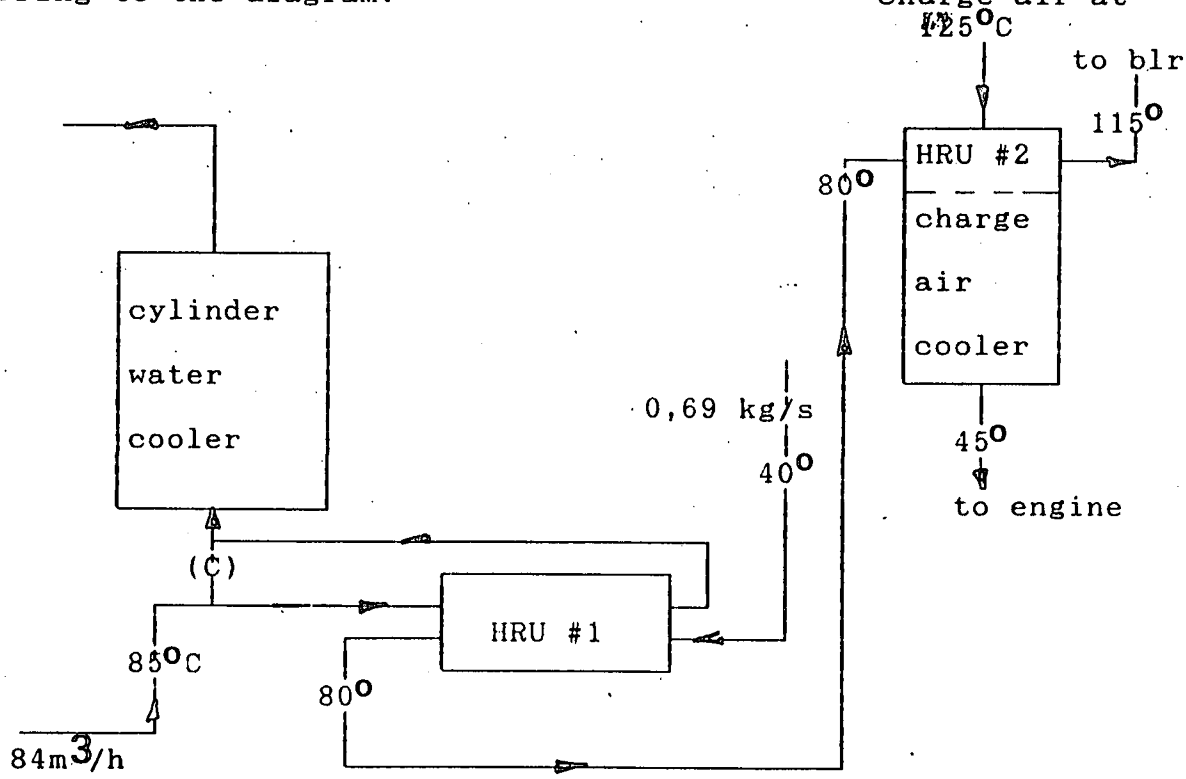
$442 = \{84 * 973{,}86 * 4{,}1940 * (80{,}4 - T_o)\}/3600$

$T_o = 75{,}7^{o}C$

Appendix: A.4.16

Design of a Feed Water System Using Heat Recovered from the Cylinder Water and Charge Air Cooler.

Referring to the diagram:



Assuming that volume flow through the pipe (C) is 3x the volume flow through HRU #1, then $Q_c = 3Q_h$

Thus $Q_c + Q_h = 0{,}0233$ m /s (84 m^3/h)

$4Q_h = 0{,}0233$

$Q_h = 0{,}0058$ m /s

From Q = V * A (Assuming the fluid velocity of 3 m/s)

$0{,}0058 = 3 * (\pi/4) * d_h$

$d_h = 49{,}6$ mm

therefore $Q_c = 0{,}0175\ m^3/s = 63\ m^3/h$

where: Q_h = volume flow through HTU #1

Q_c = volume flow through pipe (C)

d_h = HTU #1 feed pipe diameter

d_c = pipe C feed pipe diameter

The boiler water flow requirements at full boiler load = 0,69 kg/s

Thus: boiler water temperature rise over HRU #1:

$$_t = 40^\circ -- 80^\circ C$$

from $Q_{bw} = M_{bw}C_{pbw}\Delta_{tbw}$ reference $60^\circ C$

$= 0{,}69 * 4{,}1773 * 40$ $= 985{,}5\ kg/m^3$

$C_p = 4{,}1773\ kJ/kgK$

$= 115{,}3\ kW$ (Source: Appendix D.1.4)

Temperature drop of cylinder water in HRU #1

$Q_{cw} = M_{cw}C_{pcw}\Delta_{tcw}$ reference $85^\circ C$

$= 970{,}7\ kg/m^3$

$C_p = 4{,}1986\ kJ/kgK$

$= \{21 * 970{,}7 * 4{,}1986 * (85 - T_o)\}/3600$

$T_o = 80{,}2^\circ C$

The fluid temperature at inlet to the FW Distiller after mixing:

$$(84 * T_m) = (21 * 80{,}2) + (63 * 85)$$

$$T_m = 83{,}8^\circ C$$

Extracting 450 kW for FW Distiller operation, assuming that water is at full flow, then T_o (cylinder water oulet temperature)

$$Q_{fw} = M_{fw}C_{pfw}\Delta_{tfw}$$

$$450 = \{84 * 971{,}4 * 4{,}1976 * (83{,}8 - T_o)\}/3600$$

$$T_o = 79{,}1^\circ C$$

The inlet to the engine must be $75{,}8^\circ C$ ie $_t = 3{,}3^\circ C$

Thus the heat still to be dissipated: reference 77,5 C

$= 975{,}43\ kg/m^3$

$C_p = 4{,}1921\ kJ/kgK$

(Source: Appendix: D.1.4)

$$Q_d = M_d C_{pd} \Delta t_d$$
$$= (84 * 975,43 * 4,1921 * 3,3)/3600$$
$$Q_d = 314,9 \text{ kW}$$

Thus the temperature at entry into the high temperature section if the Charge Air cooler is 80°C. The charge air cooler air entry temperature is approximately 125°C. the required outlet temperature of the feed water is 115°C. Therefore, the heat balance for the high temperature section is:

$$Q_{ht} = M_{ht} C_{pht} \Delta t_{ht}$$
$$= 0,69 * 4,2108 * 35$$
$$= 101,7 \text{ kW}$$

reference 97,5°C
ρ = 962,3 kg/m³
C_p = 4,2108 kJ/kgK
(Source: Appendix D.1.4)

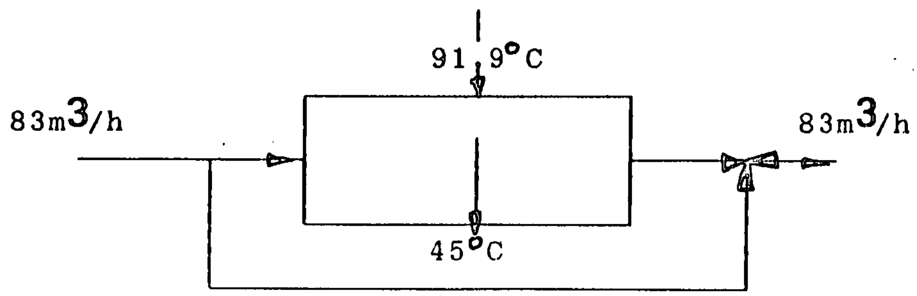
Referring to the air side: Air mass flow at 60% load is approximately 3,69 kg/s.

Thus the air temperature leaving the high temperature section is:

$$Q_a = M_a C_{pa} \Delta t_a$$
$$101,7 = 3,69 * 1,016 * (125 - T_{ao})$$
$$T_{ao} = 91,9°C$$

Now the cooling water mass flow is 83 m³/hour (sea water) and the inlet air temperature to the scavenge trunk must be 45°C.

A bypass section of the ccoling water must be fitted to control the air temperature, thus:

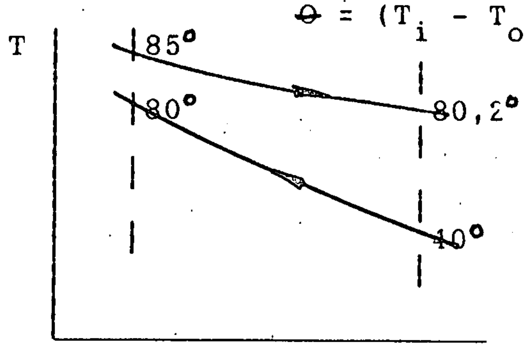


The controlling point will be the 45°C a r trunk.

Design of HRU #1

Assuming a PHE because of space considerations:

$$\theta = (T_i - T_o)/LMTD \qquad \text{-- (3-27)}$$



$O_1 = 5^\circ$
$O_2 = 40,2^\circ$

$LMTD = 16,9^\circ C \qquad \text{--(3-14)}$

$$\theta = (80 - 40)/16,9 = 2,368$$

$$\text{also } \theta = (U_d 2A)/MC_p \qquad \text{-- (3-27)}$$

where U_d is assumed at 4500 w/m^2K and 2A is due to both sides of the plate being effective.

$$2,368 = (4,5 * 2A)/(0,69 * 4,1986$$
$$= 0,762\ m^2$$

Reference Heat Exchanger Pg 72 and pg 78

Model PO Connection 42 mm

Max. heating surface area 2,98 m^2

HSA/plate 0,032 m^2

therefore, number of plates required = 24

Model A3 Connection 23 mm

Max. heating surface area 16,8 m^2 (floor)

5,8 m^2 (wall)

HSA/plate 0,058 m^2

therefore, number of plates required = 13

For the HRU #2

Fo a mass flow of 0,69 kg/s and taking a row of tubes as a pass, with the tube dimensions as follows:

ID = 19 mm

OD = 21 mm

length = 950 mm

6 tubes per row.

$$m = \rho A v$$

$$0{,}69 = 962{,}3 * (/4) * 0{,}0192 * 6 * v$$

$$v = 0{,}42 \text{ m/sec}$$

It is necessary to confirm whether the fluid flow will be turbulent: thus $Re = (\rho v d)/\mu$ reference 97,5°C

ρ = 962,3 kg/m³

u = 344 * 10^{-6} kgms

$$Re = (962{,}3 * 0{,}42 * 0{,}019)/344 * 10^{-6}$$

$$= 22402$$

Confirming turbulent flow just out of the transitional region.

To facilitate the prelimenary calculation, the Overall Coefficient will be estimated from Appendix A.4.5 Overall Heat Transfer Coefficients. Thus the average for water to compressed air is 113,5 w/m²K.

Assuming a temperature rise of 35°C

Thus to find the surface area

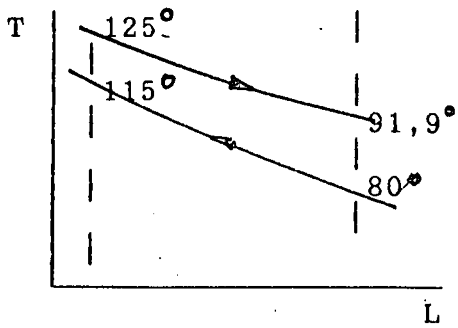
Find the minimum fluid:

MC_p (water) = 0,69 * 4,1986 = 2,897 kJ/kgK

MC_p (air) = 3,69 * 1,016 = 3,749 kJ/kgK

Thus the water is the minimum fluid and

$$\theta = (t_i - t_o)/LMTD \qquad (3\text{-}27)$$



$O_1 = 10°$
$O_2 = 11{,}9°$

$$LMTD = 10{,}92°C \qquad (3\text{-}14)$$

$$\theta = (t_i - t_o)/LMTD \qquad (3\text{-}27)$$
$$= (115 - 80)/10{,}92$$
$$= 3{,}2$$

also from equation (3-27) $\theta = (kA)/(MC_p)$

$$3{,}2 = (0{,}1135 * A)/(0{,}69 * 4{,}1986)$$
$$A = 81{,}68\ m^2$$

Using a tube of 21 mm OD and a length of 950 mm:

$$A = \pi DLn$$
$$81{,}68 = \pi * 0{,}021 * 0{,}95 * n$$
$$n = 1292 \text{ tubes.}$$

Clearly, this heat exchanger will be too large and the existing Charge air cooler only has 220 tubes. Thus the temperature rise required is too great and coupled with the low mass flow of air makes the HRU #2 in the case of the Roxana bank a poor choice for heat recovery.

Appendix: B.5.1

Blended fuel analysis for Irvin and Johnson - Cape Town

M bil Oil Southern Africa (Pty) Ltd
(REG NO 04/00909/07)

MOBIL COURT
THIBAULT SQUARE
P.O. BOX 21
8000 CAPE TOWN
REPUBLIC OF SOUTH AFRICA
TELEPHONE (021) 403-4911
TEL. ADDRESS "MOBILCAPE"
TELEX 5-26349 SA

13 March 1989

Irvin & Johnson Limited
P O Box 7444
ROGGEBAAI
8012

Attention: Mr R Haste

Dear Sir,

Please find below analysis results of the 450 second fuel blended into No 2 tank at our Cape Town Docks Depot.

Lab Sample No	273
Date sample drawn	17.02.89
Density @ 20 C kl/l	0,9628
Kinematic viscosity @ 50 C - cSt	66,28
Flash Point C	73
Water content % v/v	0,5
Carbon residue - Conradson % m/m	15,0
Sulphur % m/m	2,9
Vanadium content p.p.m.	195
Sodium content p.p.m.	24
Ash % m/m	0,05
Percentage Marine Fuel Oil	87,4
Percentage Marine Gas Oil	12,6

Yours Faithfully

J R Hey

J R HEY
Technical Advisor

c.c. Mr D Thorpe

Appendix: B.5.2

Heat Dissipation - Existing Conditions - Roxana Bank

(All reference temperatures are from Appendix: D.1.4)

Using the temperature program at 60% engine load.

eng in	eng out	Dist in	Dist out	HE in	HE out	Δ_t eng
°C	°C	°C	°C	°C	°C	°C
62	69	69	64	64	40	7

Heat dissipated in #1 FW Distiller:

reference 65,5 C ρ = 981,73 kg/m^3

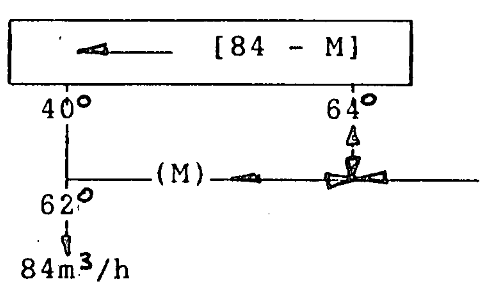
C_p = 4,1854 kJ/kg K

$$Q_{(\#1FWD)} = M_{jw} C_{pjw} \Delta_{tjw} \qquad (\Delta_{tjw} = 5°C)$$

$$= \{84 * 981,73 * 4,1854 * 5\}/3600$$

479,4 kW -- (A-1)

For mass flow through cylinder cooler:



$(84 * 62) = \{(84 - M) * 40\} + (M * 64)$

M = 77 m^3/hour

Therefore, flow through Cylinder cooler = 84 - 77 = 7m^3/h

(A-2)

Heat dissipated by cylinder water cooler

reference 52°C ρ = 989,1 kg/m^3

C_p = 4,1776 kJ/kg K

$$Q_{jw} = M_{jw}C_{pjw}\Delta_{tjw}$$

$$= \{7 * 989,1 * 4,1776 * (64 - 40)\}/3600$$

$$Q_{jw} = 192,8 \text{ kW} \qquad \text{(A-3)}$$

Confirming this result Δ_t(engine) = 7°C

reference 65,5°C ρ = 982,3 kg/m^3

C_p = 4,1845 kJ/kgK

$$Q_{eng} = M_{eng}C_{peng}\Delta_{teng}$$

$$= \{88 * 982,3 * 4,1845 * (69 - 62)\}/3600$$

$$= 703,3 \text{ kW} \qquad \text{(A-4)}$$

Heat dissipated in #1 FW Distiller and Jacket water cooler

$$Q_{tot} = 479,4 + 192,8$$

$$= 672,2 \text{ kW}$$

Add in pumping power of circulating pump of 20 kW

$$Q_{tot} = 692,2 \text{ kW} \qquad \text{(A-5)}$$

Cross checking the results with the sea water flow of 87 m^3/hour and having verified full flow through the heat exchanger.

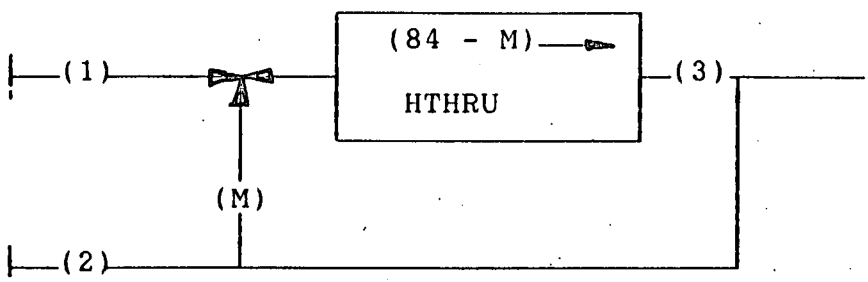
$$Q_{sw} = M_{sw}C_{psw}\Delta_{tsw}$$

$$= \{87 * 1025 * 3,998 * (40 - 33)\}/3600$$

$$693,2 \text{ kW} \qquad \text{(A-6)}$$

Appendix: B.5.3

Jacket Temperatures and Fluid Flows



(All reference conditions are taken from Appendix: D.1.4)

The flows of the fluids through the heat exchanger and the bypass line are calculated from the equation:

$(84 * 75) = \{(84 - M) * 73\} + (M * T_i)$

where T_i is the cooling water inlet temperature to the engine over a range from 85°C to 75°C.

The cooling water mass flow is assumed constant for all applications and the heat load of the heat exchanger is found from the equation:

$$Q = MC_p\Delta_t$$

The table is drawn up using 1° intervals of temperature drop.

Appendix: B.5.4

Mass flows and heat loads - Cylinder Cooling Water

	unit	cond 1	cond 2	cond 3	cond 4	cond 5
mass flow (tot)	kg/s	23,07	23,07	23,07	23,07	23,07
mass flow (HE)	kg/s	4,2	4,6	2,9	5,5	nil
cyl water in (dist)	°C	68	69	68,9	72,9	62,9
cyl water out (dist)	°C	64	65	64	68,9	60
cyl water in (HE)	°C	64	65	64	68,9	no
cyl water out (HE)	°C	42	40	40	40	flow
cyl water in (engine)	°C	60	60	61	62	60
heat load (HE)	kW	386	480	289	666	nil
heat load (dist)	kW	386	386	472	386	280
heat load (total)	kW	772	866	761	1051	280

condition 1: vessel steaming only

condition 2: vessel trawling, fish meal plant, factory on

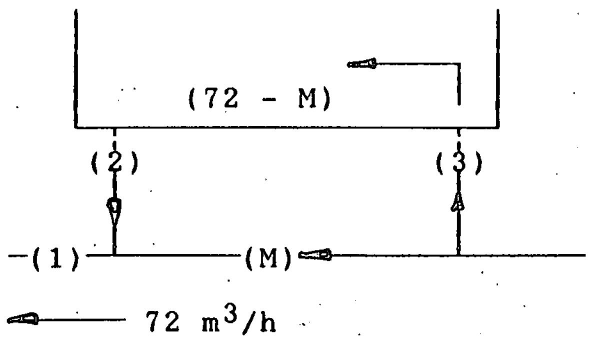
condition 3: vessel trawling, fish meal plant, factory off

condition 4: vessel steaming, fish meal plant, factory on

condition 5: vessel drifting, fish meal plant, factory off

Appendix: B.5.5

Lubricating Oil Temperatures and Fluid Flows



All reference conditions are taken from Appendix: D.1.4)

The fluid flows through the heat exchanger are calculated from the equation:

$$[72 * (1)] = \{[72 - M] * (2)\} + [M * (3)]$$

The lubricating oil mass flow is assumed to be constant for all applications and the heat load of the heat exchanger is found from the equation:

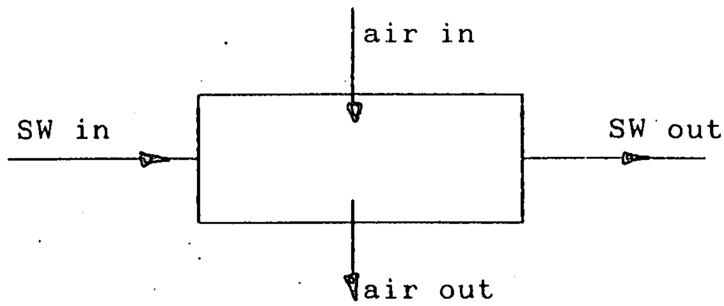
$$Q = Mc_p\Delta t$$

	units	cond 1	cond 2	cond 3	cond 4	cond 5
mass flow (tot)	kg/s	17,2	17,2	17,2	17,2	17,2
mass flow (HE)	kg/s	14,2	11,9	12,9	14,1	15,3
L.O. in (HE)	^oC(3)	65	66	65	65	62
L.O. out (HE)	^oC(2)	52,5	53	53	54	53
L.O. in (Eng)	^oC(1)	55	57	56	56	54
heat load (HE)	kW	357	311	311	311	276

Condition 1: vessel steaming only
Condition 2: vessel trawling, fish meal plant, factory on
Condition 3: vessel trawling, fish meal plant, factory off
Condition 4: vessel steaming, fish meal plant, factory on
Condition 5: vessel drifting, fish meal plant, factory off

Appendix: B.5.6

Charge Air Temperatures and Heat Loads



The heat load can be calculated from the equation:

$Q = MC_p\Delta t$ using the air mass flow or the sea water mass flow. There is full flow through the charge air cooler of air and sea water at all times.

	units	cond 1	cond 2	cond 3	cond 4	cond 5
mass flow (SW)	kg/s	24,94	24,94	24,94	24,94	24,94
mass flow (AIR)	kg/s	4,18	4,18	3,95	4,26	3,50
air in	^oC	130	135	100	120	90
air out	^oC	37	37	36	39	34
sea water in	^oC	18	18	18	18	18
sea water out	^oC	25	26	24	25	23
heat load (HE)	kW	689	788	598	696	498

Condition 1: vessel steaming only

Condition 2: vessel trawling, fish meal plant, factory on

Condition 3: vessel trawling, fish meal plant, factory off

Condition 4: vessel steaming, fish meal plant, factory on

Condition 5: vessel drifting, fish meal plant, factory off

Appendix: C.6.1

Temperatures, Mass Flows and Heat Loads

SA Winterberg, MV Border.

		units	SA Winterberg	Border	
JACKETS	eng out	^oC	88/90	83/84	
	dist in	^oC	88	-	
	dist out	^oC	82	-	
	HE in	^oC	87	84/84	
	HE out	^oC	38	-	
	eng in	^oC	79	78/78	
	SW in	^oC	28	no	(36,1^oC)
	SW out	^oC	32	record	(44,7^oC)
PISTON	eng out	^oC	64	xx	
	HE in	^oC	64	xx	
	HE out	^oC	46	xx	
	eng in	^oC	48	xx	
	SW in	^oC	23	xx	
	SW out	^oC	38	xx	
LUBOIL	eng out	^oC	46/43	68/72	
	HE in	^oC	46/43	68/72	
	HE out	^oC	36/36	-	
	eng in	^oC	41/40	55/58	
	SW in	^oC	25/25	32	
	SW out	^oC	32/32	36,1	

SA Winterberg engine revs/min = 105 - twin screw,PTO on crankshaft

MV Border engine revs/min = 500 steady - two engines driving into gearbox with PTO

Appendix: C.6.1 continued

		units	SA Winterberg	MV Border
M A S S	F L O W			
	jacket	kg/s	98	45,5
	piston	kg/s	23,9	xx
	lub oil	kg/s	51	27,7
	sea water	kg/s	171,8	97
H E A T	L O A D			
	jkt HE	kW	4845	1000
	dist	kW	1605	xxx
	jkt total	kW	6450	1000
	pist HE	kW	411	xxx
	lub oil HE	kW	512	660
E X H	G A S			
	eng out	°C	370/365	580
	t/c in	°C	370/365	580
	t/c out	°C	308/315	395

1) SA Winterberg - Temperatures and pressures recorded 01 NOV 1989.

2) MV Border - Temperatures and pressures recorded APRIL 1989.

Appendix: C.6.2

Spreadsheet Showing the Effect of Reducing Pinch Point on Heat Transfer Area

1		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C					
3	DENSITY kg/m3	1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg	4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK	0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.	13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms	0.02	0.01	0.01	0.00	0.00
8						
9	TEMP IN (p)		°C	85.00		
10	TEMP OUT (p)		°C	83.52		
11	BULK T (p)		°C	84.26		
12	DENSITY (p)	ρ	kg/m3	971.23		
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19		
14	THERM COND (p)	k	W/mK	0.67		
15	PRANDTL NO. (p)	Pr		2.12		
16	DYNAMIC VISC (p)	μ	kg/ms	0.00		
17	REYNOLDS No (p)	Re		5.048E+5		
18	NUSSELT No (p)	Nu		1,078.54		
19	FLUID COEFF.(p)	ho	kW/m2K	8.58		
20	TEMP IN (s)		°C	43.80		
21	TEMP OUT (s)		°C	56.00		
22	DIFF T (s)		°C	49.90		
23	DENSITY (s)	ρ	kg/m3	990.10		
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18		
25	THERM COND (s)	k	W/mK	0.64		
26	PRANDTL NO. (s)	Pr		3.69		
27	DYNAMIC VISC (p)	μ	kg/ms	0.00		
28	REYNOLDS No (s)	Re		1.376E+4		
29	NUSSELT No (s)	Nu		72.68		
30	FLUID COEFF.(s)	hi	kW/m2K	7.61		
31	FLUID COEFF.CORRECTED	hio	kW/m2K	6.63		
32	RECIP CLEAN COEFF			0.29		
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.44		
34	DIRT FACTOR	Rd		0.00		
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.42		
36	θ1		°C	29.00		
37	θ2		°C	39.72		
38	θ1/θ2		°C	0.73		
39	Ln			-0.31		
40	L.M.T.D.			34.08		
41	VOLUME FLOW (p)		cm/hr	42.00		
42	VOLUME FLOW (s)		cm/hr	5.00		
43	MASS FLOW (p)		kg/s	11.33		
44	MASS FLOW (s)		kg/s	1.38		
45	HEAT RECOVERED	Q	kW	70.09		
46	SURFACE AREA	A	sq.m.	0.60		
47	TUBE DIAMETER ID		m	0.01		
48	No.of TUBES			37.00		
49	ADD 15% FOR FOULING	A	sq.m.	0.69		
50	THERMAL LENGTH	h	m	0.97		
51	FLUID VELOCITY (p)		m/s	2.09		
52	FLUID VELOCITY (s)		m/s	1.28		
53	SHELL DIAMETER ID		m	0.10		
54	EQUIV.SHELL DIA.		m	0.08		
55	PINCH POINT		°C	27.52		

Spreadsheet #2

1			0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C						
3	DENSITY kg/m3		1,002.27	1,000.50	894.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg		4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK		0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.		13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms		0.02	0.01	0.01	0.00	0.00
8							
9	TEMP IN (p)			°C	85.00		
10	TEMP OUT (p)			°C	83.28		
11	BULK T (p)			°C	84.14		
12	DENSITY (p)	ρ		kg/m3	971.31		
13	SPECIFIC HT (p)	Cp		kJ/kgK	4.19		
14	THERM COND (p)	k		W/mK	0.67		
15	PRANDTL NO. (p)	Pr			2.12		
16	DYNAMIC VISC (p)	μ		kg/ms	0.00		
17	REYNOLDS No (p)	Re			5.042E+5		
18	NUSSELT No (p)	Nu			1,078.00		
19	FLUID COEFF.(p)	ho		kW/m2K	8.58		
20	TEMP IN (s)			°C	43.80		
21	TEMP OUT (s)			°C	58.00		
22	DIFF T (s)			°C	50.90		
23	DENSITY (s)	ρ		kg/m3	989.64		
24	SPECIFIC HT (s)	Cp		kJ/kgK	4.18		
25	THERM COND (s)	k		W/mK	0.64		
26	PRANDTL NO. (s)	Pr			3.62		
27	DYNAMIC VISC (p)	μ		kg/ms	0.00		
28	REYNOLDS No (s)	Re			1.399E+4		
29	NUSSELT No (s)	Nu			73.17		
30	FLUID COEFF.(s)	hi		kW/m2K	7.68		
31	FLUID COEFF.CORRECTED	hio		kW/m2K	6.69		
32	RECIP CLEAN COEFF				0.29		
33	CLEAN COEFFICIENT	Uc		kW/m2K	3.46		
34	DIRT FACTOR	Rd			0.00		
35	DESIGN COEFFICIENT	Ud		kW/m2K	3.44		
36	i1			°C	27.00		
37	i2			°C	39.48		
38	i1/i2			°C	0.68		
39	Ln				-0.38		
40	L.M.T.D.				32.85		
41	VOLUME FLOW (p)			cm/hr	42.00		
42	VOLUME FLOW (s)			cm/hr	5.00		
43	MASS FLOW (p)			kg/s	11.33		
44	MASS FLOW (s)			kg/s	1.37		
45	HEAT RECOVERED	Q		kW	81.54		
46	SURFACE AREA	A		sq.m.	0.72		
47	TUBE DIAMETER ID			m	0.01		
48	No.of TUBES				37.00		
49	ADD 15% FOR FOULING	A		sq.m.	0.83		
50	THERMAL LENGTH	h		m	1.17		
51	FLUID VELOCITY (p)			m/s	2.09		
52	FLUID VELOCITY (s)			m/s	1.28		
53	SHELL DIAMETER ID			m	0.10		
54	EQUIV.SHELL DIA.			m	0.08		
55	PINCH POINT			°C	25.28		

Spreadsheet #3

1		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C					
3	DENSITY kg/m3	1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg	4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK	0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.	13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms	0.02	0.01	0.01	0.00	0.00
8						
9	TEMP IN (p)		°C	85.00		
10	TEMP OUT (p)		°C	83.04		
11	BULK T (p)		°C	83.90		
12	DENSITY (p)	ρ	kg/m3	971.47		
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19		
14	THERM COND (p)	k	W/mK	0.67		
15	PRANDTL NO. (p)	Pr		2.13		
16	DYNAMIC VISC (p)	μ	kg/ms	0.00		
17	REYNOLDS No (p)	Re		5.030E+5		
18	NUSSELT No (p)	Nu		1,076.92		
19	FLUID COEFF.(p)	ho	kW/m2K	8.57		
20	TEMP IN (s)		°C	43.80		
21	TEMP OUT (s)		°C	60.00		
22	DIFF T (s)		°C	51.90		
23	DENSITY (s)	ρ	kg/m3	989.19		
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18		
25	THERM COND (s)	k	W/mK	0.64		
26	PRANDTL NO. (s)	Pr		3.55		
27	DYNAMIC VISC (p)	μ	kg/ms	0.00		
28	REYNOLDS No (s)	Re		1.421E+4		
29	NUSSELT No (s)	Nu		73.67		
30	FLUID COEFF.(s)	hi	kW/m2K	7.74		
31	FLUID COEFF.CORRECTED	hio	kW/m2K	6.75		
32	RECIP CLEAN COEFF			0.29		
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.47		
34	DIRT FACTOR	Rd		0.00		
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.45		
36	*i*1		°C	25.00		
37	*i*2		°C	39.24		
38	*i*1/*i*2		°C	0.64		
39	Ln			-0.45		
40	L.M.T.D.			31.59		
41	VOLUME FLOW (p)		cm/hr	42.00		
42	VOLUME FLOW (s)		cm/hr	5.00		
43	MASS FLOW (p)		kg/s	11.33		
44	MASS FLOW (s)		kg/s	1.37		
45	HEAT RECOVERED	Q	kW	92.98		
46	SURFACE AREA	A	sq.m.	0.85		
47	TUBE DIAMETER ID		m	0.01		
48	No.of TUBES			37.00		
49	ADD 15% FOR FOULING	A	sq.m.	0.98		
50	THERMAL LENGTH	*h*	m	1.38		
51	FLUID VELOCITY (p)		m/s	2.09		
52	FLUID VELOCITY (s)		m/s	1.28		
53	SHELL DIAMETER ID		m	0.10		
54	EQUIV.SHELL DIA.		m	0.08		
55	PINCH POINT		°C	23.04		

Spreadsheet #4

#			0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
1							
2	TEMPERATURE °C						
3	DENSITY kg/m3		1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg		4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK		0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.		13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms		0.02	0.01	0.01	0.00	0.00
8							
9	TEMP IN (p)			°C	85.00		
10	TEMP OUT (p)			°C	82.80		
11	BULK T (p)			°C	83.90		
12	DENSITY (p)	ρ		kg/m3	971.47		
13	SPECIFIC HT (p)	Cp		kJ/kgK	4.19		
14	THERM COND (p)	k		W/mK	0.67		
15	PRANDTL NO. (p)	Pr			2.13		
16	DYNAMIC VISC (p)	μ		kg/ms	0.00		
17	REYNOLDS No (p)	Re			5.030E+5		
18	NUSSELT No (p)	Nu			1,076.92		
19	FLUID COEFF.(p)	ho		kW/m2K	8.57		
20	TEMP IN (s)			°C	43.80		
21	TEMP OUT (s)			°C	62.00		
22	DIFF T (s)			°C	52.90		
23	DENSITY (s)	ρ		kg/m3	988.73		
24	SPECIFIC HT (s)	Cp		kJ/kgK	4.18		
25	THERM COND (s)	k		W/mK	0.64		
26	PRANDTL NO. (s)	Pr			3.49		
27	DYNAMIC VISC (p)	μ		kg/ms	0.00		
28	REYNOLDS No (s)	Re			1.445E+4		
29	NUSSELT No (s)	Nu			74.19		
30	FLUID COEFF.(s)	hi		kW/m2K	7.81		
31	FLUID COEFF.CORRECTED	hio		kW/m2K	6.81		
32	RECIP CLEAN COEFF				0.29		
33	CLEAN COEFFICIENT	Uc		kW/m2K	3.49		
34	DIRT FACTOR	Rd			0.00		
35	DESIGN COEFFICIENT	Ud		kW/m2K	3.47		
36	θ1			°C	23.00		
37	θ2			°C	39.00		
38	θ1/θ2			°C	0.59		
39	Ln				-0.53		
40	L.M.T.D.				30.30		
41	VOLUME FLOW (p)			cm/hr	42.00		
42	VOLUME FLOW (s)			cm/hr	5.00		
43	MASS FLOW (p)			kg/s	11.34		
44	MASS FLOW (s)			kg/s	1.37		
45	HEAT RECOVERED	Q		kW	104.41		
46	SURFACE AREA	A		sq.m.	0.99		
47	TUBE DIAMETER ID			m	0.01		
48	No.of TUBES				37.00		
49	ADD 15% FOR FOULING	A		sq.m.	1.14		
50	THERMAL LENGTH	h		m	1.61		
51	FLUID VELOCITY (p)			m/s	2.09		
52	FLUID VELOCITY (s)			m/s	1.28		
53	SHELL DIAMETER ID			m	0.10		
54	EQUIV.SHELL DIA.			m	0.08		
55	PINCH POINT			°C	20.80		

Spreadsheet #5

1		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C					
3	DENSITY kg/m3	1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg	4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK	0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.	13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms	0.02	0.01	0.01	0.00	0.00
8						
9	TEMP IN (p)		°C	85.00		
10	TEMP OUT (p)		°C	82.56		
11	BULK T (p)		°C	83.66		
12	DENSITY (p)	ρ	kg/m3	971.63		
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19		
14	THERM COND (p)	k	W/mK	0.67		
15	PRANDTL NO. (p)	Pr		2.13		
16	DYNAMIC VISC (p)	μ	kg/ms	0.00		
17	REYNOLDS No (p)	Re		5.018E+5		
18	NUSSELT No (p)	Nu		1,075.85		
19	FLUID COEFF.(p)	ho	kW/m2K	8.56		
20	TEMP IN (s)		°C	43.80		
21	TEMP OUT (s)		°C	64.00		
22	DIFF T (s)		°C	53.90		
23	DENSITY (s)	ρ	kg/m3	988.28		
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18		
25	THERM COND (s)	k	W/mK	0.64		
26	PRANDTL NO. (s)	Pr		3.42		
27	DYNAMIC VISC (p)	μ	kg/ms	0.00		
28	REYNOLDS No (s)	Re		1.470E+4		
29	NUSSELT No (s)	Nu		74.72		
30	FLUID COEFF.(s)	hi	kW/m2K	7.88		
31	FLUID COEFF.CORRECTED	hio	kW/m2K	6.87		
32	RECIP CLEAN COEFF			0.29		
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.51		
34	DIRT FACTOR	Rd		0.00		
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.49		
36	θ1		°C	21.00		
37	θ2		°C	38.76		
38	θ1/θ2		°C	0.54		
39	Ln			-0.61		
40	L.M.T.D.			28.98		
41	VOLUME FLOW (p)		cm/hr	42.00		
42	VOLUME FLOW (s)		cm/hr	5.00		
43	MASS FLOW (p)		kg/s	11.34		
44	MASS FLOW (s)		kg/s	1.37		
45	HEAT RECOVERED	Q	kW	115.83		
46	SURFACE AREA	A	sq.m.	1.15		
47	TUBE DIAMETER ID		m	0.01		
48	No.of TUBES			37.00		
49	ADD 15% FOR FOULING	A	sq.m.	1.32		
50	THERMAL LENGTH	θ	m	1.86		
51	FLUID VELOCITY (p)		m/s	2.09		
52	FLUID VELOCITY (s)		m/s	1.28		
53	SHELL DIAMETER ID		m	0.10		
54	EQUIV.SHELL DIA.		m	0.08		
55	PINCH POINT		°C	18.56		

Spreadsheet #6

1			0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE ºC						
3	DENSITY kg/m3		1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg		4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK		0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.		13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms		0.02	0.01	0.01	0.00	0.00
8							
9	TEMP IN (p)			ºC	85.00		
10	TEMP OUT (p)			ºC	82.32		
11	BULK T (p)			ºC	83.54		
12	DENSITY (p)	ρ		kg/m3	971.71		
13	SPECIFIC HT (p)	Cp		kJ/kgK	4.19		
14	THERM COND (p)	k		W/mK	0.67		
15	PRANDTL NO. (p)	Pr			2.14		
16	DYNAMIC VISC (p)	μ		kg/ms	0.00		
17	REYNOLDS No (p)	Re			5.012E+5		
18	NUSSELT No (p)	Nu			1,075.32		
19	FLUID COEFF.(p)	ho		kW/m2K	8.55		
20	TEMP IN (s)			ºC	43.80		
21	TEMP OUT (s)			ºC	66.00		
22	DIFF T (s)			ºC	54.90		
23	DENSITY (s)	ρ		kg/m3	987.82		
24	SPECIFIC HT (s)	Cp		kJ/kgK	4.18		
25	THERM COND (s)	k		W/mK	0.64		
26	PRANDTL NO. (s)	Pr			3.36		
27	DYNAMIC VISC (p)	μ		kg/ms	0.00		
28	REYNOLDS No (s)	Re			1.495E+4		
29	NUSSELT No (s)	Nu			75.26		
30	FLUID COEFF.(s)	hi		kW/m2K	7.95		
31	FLUID COEFF.CORRECTED	hio		kW/m2K	6.93		
32	RECIP CLEAN COEFF				0.28		
33	CLEAN COEFFICIENT	Uc		kW/m2K	3.52		
34	DIRT FACTOR	Rd			0.00		
35	DESIGN COEFFICIENT	Ud		kW/m2K	3.51		
36	θ_1			ºC	19.00		
37	θ_2			ºC	38.52		
38	θ_1/θ_2			ºC	0.49		
39	Ln				-0.71		
40	L.M.T.D.				27.62		
41	VOLUME FLOW (p)			cm/hr	42.00		
42	VOLUME FLOW (s)			cm/hr	5.00		
43	MASS FLOW (p)			kg/s	11.34		
44	MASS FLOW (s)			kg/s	1.37		
45	HEAT RECOVERED	Q		kW	127.24		
46	SURFACE AREA	A		sq.m.	1.31		
47	TUBE DIAMETER ID			m	0.01		
48	No.of TUBES				37.00		
49	ADD 15% FOR FOULING	A		sq.m.	1.51		
50	THERMAL LENGTH			m	2.13		
51	FLUID VELOCITY (p)			m/s	2.09		
52	FLUID VELOCITY (s)			m/s	1.28		
53	SHELL DIAMETER ID			m	0.10		
54	EQUIV.SHELL DIA.			m	0.08		
55	PINCH POINT			ºC	16.32		

Spreadsheet #7

1			0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C						
3	DENSITY kg/m3		1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg		4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK		0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.		13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms		0.02	0.01	0.01	0.00	0.00
8							
9	TEMP IN (p)			°C	85.00		
10	TEMP OUT (p)			°C	82.08		
11	BULK T (p)			°C	83.36		
12	DENSITY (p)		ρ	kg/m3	971.83		
13	SPECIFIC HT (p)		Cp	kJ/kgK	4.19		
14	THERM COND (p)		k	W/mK	0.67		
15	PRANDTL NO. (p)		Pr		2.14		
16	DYNAMIC VISC (p)		μ	kg/ms	0.00		
17	REYNOLDS No (p)		Re		5.003E+5		
18	NUSSELT No (p)		Nu		1,074.52		
19	FLUID COEFF.(p)		ho	kW/m2K	8.54		
20	TEMP IN (s)			°C	43.80		
21	TEMP OUT (s)			°C	68.00		
22	DIFF T (s)			°C	55.90		
23	DENSITY (s)		ρ	kg/m3	987.37		
24	SPECIFIC HT (s)		Cp	kJ/kgK	4.18		
25	THERM COND (s)		k	W/mK	0.65		
26	PRANDTL NO. (s)		Pr		3.29		
27	DYNAMIC VISC (p)		μ	kg/ms	0.00		
28	REYNOLDS No (s)		Re		1.521E+4		
29	NUSSELT No (s)		Nu		75.81		
30	FLUID COEFF.(s)		hi	kW/m2K	8.02		
31	FLUID COEFF.CORRECTED		hio	kW/m2K	6.99		
32	RECIP CLEAN COEFF				0.28		
33	CLEAN COEFFICIENT		Uc	kW/m2K	3.54		
34	DIRT FACTOR		Rd		0.00		
35	DESIGN COEFFICIENT		Ud	kW/m2K	3.52		
36	θ_1			°C	17.00		
37	θ_2			°C	38.28		
38	θ_1/θ_2			°C	0.44		
39	Ln				-0.81		
40	L.M.T.D.				26.22		
41	VOLUME FLOW (p)			cm/hr	42.00		
42	VOLUME FLOW (s)			cm/hr	5.00		
43	MASS FLOW (p)			kg/s	11.34		
44	MASS FLOW (s)			kg/s	1.37		
45	HEAT RECOVERED		Q	kW	138.64		
46	SURFACE AREA		A	sq.m.	1.50		
47	TUBE DIAMETER ID			m	0.01		
48	No.of TUBES				37.00		
49	ADD 15% FOR FOULING		A	sq.m.	1.73		
50	THERMAL LENGTH		θ	m	2.43		
51	FLUID VELOCITY (p)			m/s	2.09		
52	FLUID VELOCITY (s)			m/s	1.28		
53	SHELL DIAMETER ID			m	0.10		
54	EQUIV.SHELL DIA.			m	0.08		
55	PINCH POINT			°C	14.08		

Spreadsheet #8

1		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C					
3	DENSITY kg/m3	1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg	4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK	0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.	13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms	0.02	0.01	0.01	0.00	0.00
8						
9	TEMP IN (p)		°C	85.00		
10	TEMP OUT (p)		°C	81.72		
11	BULK T (p)		°C	83.36		
12	DENSITY (p)	ρ	kg/m3	971.83		
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19		
14	THERM COND (p)	k	W/mK	0.67		
15	PRANDTL NO. (p)	Pr		2.14		
16	DYNAMIC VISC (p)	μ	kg/ms	0.00		
17	REYNOLDS No (p)	Re		5.003E+5		
18	NUSSELT No (p)	Nu		1,074.52		
19	FLUID COEFF.(p)	ho	kW/m2K	8.54		
20	TEMP IN (s)		°C	43.80		
21	TEMP OUT (s)		°C	71.00		
22	DIFF T (s)		°C	57.40		
23	DENSITY (s)	ρ	kg/m3	986.68		
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18		
25	THERM COND (s)	k	W/mK	0.65		
26	PRANDTL NO. (s)	Pr		3.19		
27	DYNAMIC VISC (p)	μ	kg/ms	0.00		
28	REYNOLDS No (s)	Re		1.562E+4		
29	NUSSELT No (s)	Nu		76.66		
30	FLUID COEFF.(s)	hi	kW/m2K	8.14		
31	FLUID COEFF.CORRECTED	hio	kW/m2K	7.09		
32	RECIP CLEAN COEFF			0.28		
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.57		
34	DIRT FACTOR	Rd		0.00		
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.55		
36	θ1		°C	14.00		
37	θ2		°C	37.92		
38	θ1/θ2		°C	0.37		
39	Ln			-1.00		
40	L.M.T.D.			24.01		
41	VOLUME FLOW (p)		cm/hr	42.00		
42	VOLUME FLOW (s)		cm/hr	5.00		
43	MASS FLOW (p)		kg/s	11.34		
44	MASS FLOW (s)		kg/s	1.37		
45	HEAT RECOVERED	Q	kW	155.71		
46	SURFACE AREA	A	sq.m.	1.83		
47	TUBE DIAMETER ID		m	0.01		
48	No.of TUBES			37.00		
49	ADD 15% FOR FOULING	A	sq.m.	2.10		
50	THERMAL LENGTH	ℓ	m	2.96		
51	FLUID VELOCITY (p)		m/s	2.09		
52	FLUID VELOCITY (s)		m/s	1.28		
53	SHELL DIAMETER ID		m	0.10		
54	EQUIV.SHELL DIA.		m	0.08		
55	PINCH POINT		°C	10.72		

Spreadsheet #9

		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
1						
2	TEMPERATURE °C					
3	DENSITY kg/m3	1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg	4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK	0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.	13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms	0.02	0.01	0.01	0.00	0.00

8				
9	TEMP IN (p)		°C	85.00
10	TEMP OUT (p)		°C	81.48
11	BULK T (p)		°C	83.24
12	DENSITY (p)	ρ	kg/m3	971.92
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19
14	THERM COND (p)	k	W/mK	0.67
15	PRANDTL NO. (p)	Pr		2.14
16	DYNAMIC VISC (p)	μ	kg/ms	0.00
17	REYNOLDS No (p)	Re		4.997E+5
18	NUSSELT No (p)	Nu		1,073.99
19	FLUID COEFF.(p)	ho	kW/m2K	8.54
20	TEMP IN (s)		°C	43.80
21	TEMP OUT (s)		°C	73.00
22	DIFF T (s)		°C	58.40
23	DENSITY (s)	ρ	kg/m3	986.23
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18
25	THERM COND (s)	k	W/mK	0.65
26	PRANDTL NO. (s)	Pr		3.13
27	DYNAMIC VISC (p)	μ	kg/ms	0.00
28	REYNOLDS No (s)	Re		1.591E+4
29	NUSSELT No (s)	Nu		77.25
30	FLUID COEFF.(s)	hi	kW/m2K	8.21
31	FLUID COEFF.CORRECTED	hio	kW/m2K	7.16
32	RECIP CLEAN COEFF			0.28
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.59
34	DIRT FACTOR	Rd		0.00
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.57
36	θ1		°C	12.00
37	θ2		°C	37.68
38	θ1/θ2		°C	0.32
39	Ln			-1.14
40	L.M.T.D.			22.44
41	VOLUME FLOW (p)		cm/hr	42.00
42	VOLUME FLOW (s)		cm/hr	5.00
43	MASS FLOW (p)		kg/s	11.34
44	MASS FLOW (s)		kg/s	1.37
45	HEAT RECOVERED	Q	kW	167.08
46	SURFACE AREA	A	sq.m.	2.09
47	TUBE DIAMETER ID		m	0.01
48	No.of TUBES			37.00
49	ADD 15% FOR FOULING	A	sq.m.	2.40
50	THERMAL LENGTH	θ	m	3.38
51	FLUID VELOCITY (p)		m/s	2.09
52	FLUID VELOCITY (s)		m/s	1.28
53	SHELL DIAMETER ID		m	0.10
54	EQUIV.SHELL DIA.		m	0.08
55	PINCH POINT		°C	8.48

Spreadsheet #10

1		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C					
3	DENSITY kg/m3	1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg	4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK	0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.	13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms	0.02	0.01	0.01	0.00	0.00
8						
9	TEMP IN (p)		°C	85.00		
10	TEMP OUT (p)		°C	81.24		
11	BULK T (p)		°C	83.12		
12	DENSITY (p)	ρ	kg/m3	972.00		
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19		
14	THERM COND (p)	k	W/mK	0.67		
15	PRANDTL NO. (p)	Pr		2.15		
16	DYNAMIC VISC (p)	μ	kg/ms	0.00		
17	REYNOLDS No (p)	Re		4.992E+5		
18	NUSSELT No (p)	Nu		1,073.46		
19	FLUID COEFF.(p)	ho	kW/m2K	8.53		
20	TEMP IN (s)		°C	43.80		
21	TEMP OUT (s)		°C	75.00		
22	DIFF T (s)		°C	59.40		
23	DENSITY (s)	ρ	kg/m3	985.77		
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18		
25	THERM COND (s)	k	W/mK	0.65		
26	PRANDTL NO. (s)	Pr		3.06		
27	DYNAMIC VISC (p)	μ	kg/ms	0.00		
28	REYNOLDS No (s)	Re		1.621E+4		
29	NUSSELT No (s)	Nu		77.85		
30	FLUID COEFF.(s)	hi	kW/m2K	8.29		
31	FLUID COEFF.CORRECTED	hio	kW/m2K	7.22		
32	RECIP CLEAN COEFF			0.28		
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.61		
34	DIRT FACTOR	Rd		0.00		
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.59		
36	θ_1		°C	10.00		
37	θ_2		°C	37.44		
38	θ_1/θ_2		°C	0.27		
39	Ln			-1.32		
40	L.M.T.D.			20.79		
41	VOLUME FLOW (p)		cm/hr	42.00		
42	VOLUME FLOW (s)		cm/hr	5.00		
43	MASS FLOW (p)		kg/s	11.34		
44	MASS FLOW (s)		kg/s	1.37		
45	HEAT RECOVERED	Q	kW,	178.44		
46	SURFACE AREA	A	sq.m.	2.39		
47	TUBE DIAMETER ID		m	0.01		
48	No.of TUBES			37.00		
49	ADD 15% FOR FOULING	A	sq.m.	2.75		
50	THERMAL LENGTH		m	3.88		
51	FLUID VELOCITY (p)		m/s	2.09		
52	FLUID VELOCITY (s)		m/s	1.28		
53	SHELL DIAMETER ID		m	0.10		
54	EQUIV.SHELL DIA.		m	0.08		
55	PINCH POINT		°C	6.24		

Spreadsheet #11

1			0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE xC						
3	DENSITY kg/m3		1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg		4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK		0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.		13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms		0.02	0.01	0.01	0.00	0.00
8							

9	TEMP IN (p)		xC	85.00
10	TEMP OUT (p)		xC	81.01
11	BULK T (p)		xC	83.00
12	DENSITY (p)	*b*	kg/m3	972.08
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19
14	THERM COND (p)	k	W/mK	0.67
15	PRANDTL NO. (p)	Pr		2.15
16	DYNAMIC VISC (p)	*f*	kg/ms	0.00
17	REYNOLDS No (p)	Re		4.986E+5
18	NUSSELT No (p)	Nu		1,072.93
19	FLUID COEFF.(p)	ho	kW/m2K	8.53
20	TEMP IN (s)		xC	43.80
21	TEMP OUT (s)		xC	77.00
22	DIFF T (s)		xC	60.40
23	DENSITY (s)	*b*	kg/m3	985.32
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18
25	THERM COND (s)	k	W/mK	0.65
26	PRANDTL NO. (s)	Pr		2.99
27	DYNAMIC VISC (p)	*f*	kg/ms	0.00
28	REYNOLDS No (s)	Re		1.652E+4
29	NUSSELT No (s)	Nu		78.47
30	FLUID COEFF.(s)	hi	kW/m2K	8.37
31	FLUID COEFF.CORRECTED	hio	kW/m2K	7.30
32	RECIP CLEAN COEFF			0.28
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.63
34	DIRT FACTOR	Rd		0.00
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.61
36	*i*1		xC	8.00
37	*i*2		xC	37.21
38	*i*1/*i*2		xC	0.22
39	Ln			-1.54
40	L.M.T.D.			19.00
41	VOLUME FLOW (p)		cm/hr	42.00
42	VOLUME FLOW (s)		cm/hr	5.00
43	MASS FLOW (p)		kg/s	11.34
44	MASS FLOW (s)		kg/s	1.37
45	HEAT RECOVERED	Q	kW	189.79
46	SURFACE AREA	A	sq.m.	2.77
47	TUBE DIAMETER ID		m	0.01
48	No.of TUBES			37.00
49	ADD 15% FOR FOULING	A	sq.m.	3.19
50	THERMAL LENGTH	*h*	m	4.49
51	FLUID VELOCITY (p)		m/s	2.09
52	FLUID VELOCITY (s)		m/s	1.28
53	SHELL DIAMETER ID		m	0.10
54	EQUIV.SHELL DIA.		m	0.08
55	PINCH POINT		xC	4.01

Spreadsheet #12

1				0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C							
3	DENSITY kg/m3			1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg			4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK			0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.			13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms			0.02	0.01	0.01	0.00	0.00
8								
9	TEMP IN (p)		°C			85.00		
10	TEMP OUT (p)		°C			80.77		
11	BULK T (p)		°C			82.96		
12	DENSITY (p)	ρ	kg/m3			972.10		
13	SPECIFIC HT (p)	Cp	kJ/kgK			4.19		
14	THERM COND (p)	k	W/mK			0.67		
15	PRANDTL NO. (p)	Pr				2.15		
16	DYNAMIC VISC (p)	μ	kg/ms			0.00		
17	REYNOLDS No (p)	Re				4.984E+5		
18	NUSSELT No (p)	Nu				1,072.75		
19	FLUID COEFF.(p)	ho	kW/m2K			8.53		
20	TEMP IN (s)		°C			43.80		
21	TEMP OUT (s)		°C			79.00		
22	DIFF T (s)		°C			61.40		
23	DENSITY (s)	ρ	kg/m3			984.86		
24	SPECIFIC HT (s)	Cp	kJ/kgK			4.18		
25	THERM COND (s)	k	W/mK			0.65		
26	PRANDTL NO. (s)	Pr				2.93		
27	DYNAMIC VISC (p)	μ	kg/ms			0.00		
28	REYNOLDS No (s)	Re				1.684E+4		
29	NUSSELT No (s)	Nu				79.11		
30	FLUID COEFF.(s)	hi	kW/m2K			8.45		
31	FLUID COEFF.CORRECTED	hio	kW/m2K			7.37		
32	RECIP CLEAN COEFF					0.27		
33	CLEAN COEFFICIENT	Uc	kW/m2K			3.65		
34	DIRT FACTOR	Rd				0.00		
35	DESIGN COEFFICIENT	Ud	kW/m2K			3.63		
36	θ1		°C			6.00		
37	θ2		°C			36.97		
38	θ1/θ2		°C			0.16		
39	Ln					-1.82		
40	L.M.T.D.					17.03		
41	VOLUME FLOW (p)		cm/hr			42.00		
42	VOLUME FLOW (s)		cm/hr			5.00		
43	MASS FLOW (p)		kg/s			11.34		
44	MASS FLOW (s)		kg/s			1.37		
45	HEAT RECOVERED	Q	kW			201.13		
46	SURFACE AREA	A	sq.m.			3.26		
47	TUBE DIAMETER ID		m			0.01		
48	No.of TUBES					37.00		
49	ADD 15% FOR FOULING	A	sq.m.			3.75		
50	THERMAL LENGTH	θ	m			5.28		
51	FLUID VELOCITY (p)		m/s			2.09		
52	FLUID VELOCITY (s)		m/s			1.28		
53	SHELL DIAMETER ID		m			0.10		
54	EQUIV.SHELL DIA.		m			0.08		
55	PINCH POINT		°C			1.77		

Spreadsheet #13

1		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C					
3	DENSITY kg/m3	1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg	4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK	0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.	13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms	0.02	0.01	0.01	0.00	0.00
8						
9	TEMP IN (p)		°C	85.00		
10	TEMP OUT (p)		°C	80.53		
11	BULK T (p)		°C	82.77		
12	DENSITY (p)	ρ	kg/m3	972.24		
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19		
14	THERM COND (p)	k	W/mK	0.67		
15	PRANDTL NO. (p)	Pr		2.15		
16	DYNAMIC VISC (p)	μ	kg/ms	0.00		
17	REYNOLDS No (p)	Re		4.974E+5		
18	NUSSELT No (p)	Nu		1,071.89		
19	FLUID COEFF.(p)	ho	kW/m2K	8.52		
20	TEMP IN (s)		°C	43.80		
21	TEMP OUT (s)		°C	81.00		
22	DIFF T (s)		°C	62.40		
23	DENSITY (s)	ρ	kg/m3	984.41		
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18		
25	THERM COND (s)	k	W/mK	0.65		
26	PRANDTL NO. (s)	Pr		2.86		
27	DYNAMIC VISC (p)	μ	kg/ms	0.00		
28	REYNOLDS No (s)	Re		1.718E+4		
29	NUSSELT No (s)	Nu		79.76		
30	FLUID COEFF.(s)	hi	kW/m2K	8.54		
31	FLUID COEFF.CORRECTED	hio	kW/m2K	7.44		
32	RECIP CLEAN COEFF			0.27		
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.66		
34	DIRT FACTOR	Rd		0.00		
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.64		
36	θ1		°C	4.00		
37	θ2		°C	36.73		
38	θ1/θ2		°C	0.11		
39	Ln			-2.22		
40	L.M.T.D.			14.76		
41	VOLUME FLOW (p)		cm/hr	42.00		
42	VOLUME FLOW (s)		cm/hr	5.00		
43	MASS FLOW (p)		kg/s	11.34		
44	MASS FLOW (s)		kg/s	1.37		
45	HEAT RECOVERED	Q	kW	212.46		
46	SURFACE AREA	A	sq.m.	3.95		
47	TUBE DIAMETER ID		m	0.01		
48	No.of TUBES			37.00		
49	ADD 15% FOR FOULING	A	sq.m.	4.54		
50	THERMAL LENGTH	θ	m	6.40		
51	FLUID VELOCITY (p)		m/s	2.09		
52	FLUID VELOCITY (s)		m/s	1.28		
53	SHELL DIAMETER ID		m	0.10		
54	EQUIV.SHELL DIA.		m	0.08		
55	PINCH POINT		°C	-0.47		

Spreadsheet #14

		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
1		0 to 20	20 to 40	40 to 60	60 to 80	80 to 100
2	TEMPERATURE °C					
3	DENSITY kg/m3	1,002.27	1,000.50	994.60	985.50	974.10
4	SPECIFIC HEAT kJ/kg	4.21	4.18	4.18	4.18	4.19
5	THERM.COND.(k) W/mK	0.55	0.60	0.63	0.65	0.67
6	PRANDTL No.	13.60	7.02	4.34	3.02	2.22
7	DYNAMIC VISC. kg/ms	0.02	0.01	0.01	0.00	0.00
8						
9	TEMP IN (p)		°C	85.00		
10	TEMP OUT (p)		°C	80.29		
11	BULK T (p)		°C	82.65		
12	DENSITY (p)	ρ	kg/m3	972.32		
13	SPECIFIC HT (p)	Cp	kJ/kgK	4.19		
14	THERM COND (p)	k	W/mK	0.67		
15	PRANDTL NO. (p)	Pr		2.16		
16	DYNAMIC VISC (p)	μ	kg/ms	0.00		
17	REYNOLDS No (p)	Re		4.968E+5		
18	NUSSELT No (p)	Nu		1,071.36		
19	FLUID COEFF.(p)	ho	kW/m2K	8.51		
20	TEMP IN (s)		°C	43.80		
21	TEMP OUT (s)		°C	83.00		
22	DIFF T (s)		°C	63.40		
23	DENSITY (s)	ρ	kg/m3	983.95		
24	SPECIFIC HT (s)	Cp	kJ/kgK	4.18		
25	THERM COND (s)	k	W/mK	0.65		
26	PRANDTL NO. (s)	Pr		2.80		
27	DYNAMIC VISC (p)	μ	kg/ms	0.00		
28	REYNOLDS No (s)	Re		1.753E+4		
29	NUSSELT No (s)	Nu		80.43		
30	FLUID COEFF.(s)	hi	kW/m2K	8.63		
31	FLUID COEFF.CORRECTED	hio	kW/m2K	7.52		
32	RECIP CLEAN COEFF			0.27		
33	CLEAN COEFFICIENT	Uc	kW/m2K	3.68		
34	DIRT FACTOR	Rd		0.00		
35	DESIGN COEFFICIENT	Ud	kW/m2K	3.66		
36	θ1		°C	2.00		
37	θ2		°C	36.49		
38	θ1/θ2		°C	0.05		
39	Ln			-2.90		
40	L.M.T.D.			11.88		
41	VOLUME FLOW (p)		cm/hr	42.00		
42	VOLUME FLOW (s)		cm/hr	5.00		
43	MASS FLOW (p)		kg/s	11.34		
44	MASS FLOW (s)		kg/s	1.37		
45	HEAT RECOVERED	Q	kW	223.77		
46	SURFACE AREA	A	sq.m.	5.14		
47	TUBE DIAMETER ID		m	0.01		
48	No.of TUBES			37.00		
49	ADD 15% FOR FOULING	A	sq.m.	5.91		
50	THERMAL LENGTH	θ	m	8.34		
51	FLUID VELOCITY (p)		m/s	2.09		
52	FLUID VELOCITY (s)		m/s	1.28		
53	SHELL DIAMETER ID		m	0.10		
54	EQUIV.SHELL DIA.		m	0.08		
55	PINCH POINT		°C	-2.71		

Equations used to run spreadsheet.

No	function
9	nil
10	nil
11	$(T_i - T_o)/2$ -- (primary fluid)
12	nil -- from appendix D.1.4
13	nil -- from appendix D.1.4
14	nil -- from appendix D.1.4
15	$(C_p u)/k$ -- (3-8c)
16	nil -- from appendix D.1.4
17.	$(\ vd)/u$ -- (3-8b)
18	$(0{,}027)Re^{0,8}Pr^{1/3}$ -- (3-13b)
19	$Nu = (h_o d)/k$ -- (3-8a)
20	nil
21	nil
22	$(t_i - t_o)$ -- (secondary fluid)
23	nil -- from appendix D.1.4
24	nil -- from appendix D.1.4
25	nil -- from appendix D.1.4
26	(3-8c)
27	nil -- from appendix D.1.4
28	(3-8b)
29	(3-13b)
30	(3-8a)
31	
32	reciprocal 31
33	(3-5)
34	(3-6a)
35	(3-6b)
36	$T_i - t_o$
37	$T_o - t_i$
38	$(T_i - t_o)/(T_o - t_i)$
39	ln
40	(3-14)
41	given
42	given
43	volume flow/sec * density
44	volume flow/sec * density
45	$Q = MC_p \ t$
46	$Q_d = U_d A \ t$
47	standard size
48	calculated - fixed
49	
50	$A = \pi dLn$
51	$Q = V * A$
52	$Q = V * A$
53	
54	(3-25)
55	$(T_o - t_o)$

Appendix: C.6.2a

Data for Heat Load vs Heat Exchanger surface area

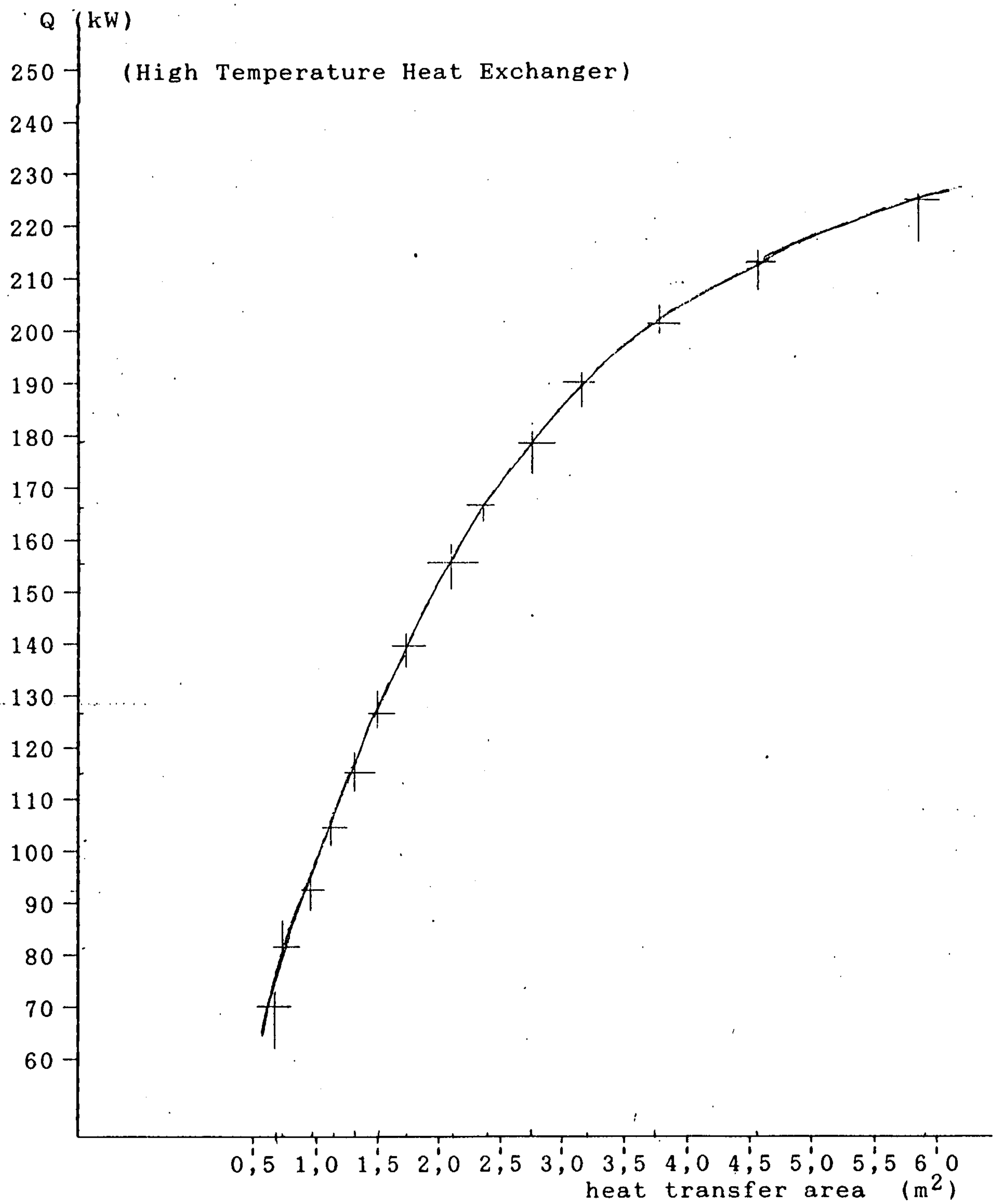
heat load Q	HE surface area
kW	m^2
70,09	0,69
81,54	0,72
92,98	0,98
104,41	1,14
115,83	1,32
127,24	1,51
138,64	1,73
155,71	2,10
167,08	2,40
178,44	2,75
189,79	3,19
201,13	3,75
212,46	4,54
223,77	5,91

NB: The calculation for the heat load against the heat transfer area is calculated for a primary fluid flow of 42m^3/hour and a secondary fluid flow of 5 m^3/hour. The selected heat transfer area must be miltiplied by 2 as must the heat recovered.

Appendix: C.6.2b

Effect of Heat Recovered to Heat Transfer Surface

with Reducing Pinch Point



Appendix: C.6.3

Vessel Trial Data: Roxana Bank

date: 25/02/78	cond 1	cond 2	
time/	16h30	18h00	
load	75%	100%	
rev/min	500	500	
air in HE	75°C	83°C	
air out HE/press	35°C/123	35°C/170	(Air press kPa)
lub oil press	550kPa	540kPa	
lub oil in HE/eng	55/45°C	47/54°C	
lub oil out HE	36°C	37°C	
cyl water press	265kPa	265kPa	
cyl water in HE	64°C	58°C	
cyl water out HE	28°C	30°C	
cyl water eng in	60°C	58°C	
SW press (system)	260kPa	260kPa	
SW/lub oil HE	24/24°C	27/30°C	
SW/cyl HE	25/29°C	29/31°C	
engineroom temp	33°C	34°C	
air temp	18°C	18°C	
SW temp	-2°C	-2°C	

(Source: Original trial data sheetJEDN B-417/12, page 31)

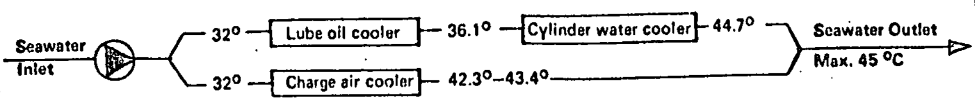
Appendix: C.6.4

Sulzer bulletin 4-107.159.310

SULZER Z 40 – Engine Marine	TECHNICAL DATA FOR ANCILLARY EQUIPMENT Based on diesel engine programme 4–107.059.310 Engine particulars: External mounted charge air cooler. Bore-cooled liner with thermal insulating tubes.	533 kW/Cyl. 530 rev/min

Cyl. dia. 400 mm – Stroke 480 mm – m. e. p. 20.02 bar – Piston speed 8.48 m/s – Piston displacement 60.3 dm^3

1.0 Cooler Data for the following sea-water arrangement and assuming a max. sea-water temperature of 32 °C



			6 ZL 40	8 ZL 40	10 ZV 40	12 ZV 40	14 ZV 40	16 ZV 40	18 ZV 40
	Max. continuous output at 530 rev/min 1)	kW	3'198	4'264	5'330	6'396	7'462	8'528	9'594
	Engine form		In-line	In-line	V-form	V-form	V-form	V-form	V-form
	Number of cylinders		6	8	10	12	14	16	18
	Number and Type of BBC turbocharger		1xVTR401	1xVTR401	1xVTR501	2xVTR401	2xVTR401	2xVTR401	2xVTR501
1.1	Oil cooler : Heat dissipation	kW	411	548	685	822	960	1'095	1'235
	Oil quantity	m3/h	72	96	110	132	154	176	198
	Oil temperature in/out	°C	66.4/55	66.4/55	67.5/55	67.5/55	67.5/55	67.5/55	67.5/55
	Sea water quantity	m3/h	87	116	145	174	203	232	261
	Sea water temperature in/out	°C	32/36.1	32/36.1	32/36.1	32/36.1	32/36.1	32/36.1	32/36.1
	Mean log. temp. difference ϑm	°C	26.4	26.4	27	27	27	27	27
1.2	Cylinder water cooler : Heat dissipation	kW	875	1'165	1'460	1'750	2'040	2'335	2'625
	Fresh water quantity	m3/h	84	112	140	168	196	224	252
	Fresh water temperature in/out	°C	85/75.8	85/75.8	85/75.8	85/75.8	85/75.8	85/75.8	85/75.8
	Sea water quantity	m3/h	87	116	145	174	203	232	261
	Sea water temperature in/out	°C	36.1/44.7	36.1/44.7	36.1/44.7	36.1/44.7	36.1/44.7	36.1/44.7	36.1/44.7
	Mean log. temp. difference ϑm	°C	40	40	40	40	40	40	40
1.3	Charge air cooler : Heat dissipation	kW	990	1'320	1'650	1'980	2'310	2'640	2'970
	Sea water quantity	m3/h	83	110	124	165	193	220	223
	Sea water temperature in/out	°C	32/42.3	32/42.3	32/43.4	32/42.3	32/42.3	32/42.3	32/43.4
2.1	Sea water pump : Capacity 2)	m3/h	170	226	269	339	396	452	484
	Total head 3)	bar	1.8	1.8	1.8	1.8	1.8	1.8	1.8
2.2	Cylinder water pump : Capacity 2)	m3/h	84	112	140	168	196	224	252
	Total head 3)	bar	2.5	2.5	2.5	2.5	2.5	2.5	2.5
2.3	Fuel valve water pump : Capacity 2)	m3/h	3	4	5	6	7	8	9
	Total head 3)	bar	3	3	3	3	3	3	3
2.4	Lube oil pump : Capacity 2)	m3/h	72	96	110	132	154	176	198
	Delivery pressure 3)	bar	6	6	6	6	6	6	6
2.5	Fuel oil booster pump : Capacity 2)	m3/h	2.4	3.2	4	4.8	5.6	6.4	7.2
	Delivery pressure	bar	10	10	10	10	10	10	10
3.1	Exhaust gases : at 45 °C ambient 4)								
	Quantity at 100 % load	kg/h	22'600	30'200	37'700	45'200	52'800	60'300	67'900
	85 % load	kg/h	19'700	25'700	32'200	39'400	44'700	51'500	59'100
	70 % load	kg/h	16'600	21'100	26'300	33'200	36'500	42'100	49'800
	Temp. turbine outlet at 100 % load	°C	400	400	400	400	400	400	400
	Temp. turbine outlet at 85 & 70 % load	°C	390/385	395/395	395/395	390/385	400/400	395/395	390/385
3.2	Exhaust gases : at 27 °C ambient 4)								
	Quantity at 100 % load	kg/h	23'800	31'800	39'700	47'700	55'600	63'500	71'500
	85 % load	kg/h	20'800	27'200	34'000	41'700	47'200	54'400	62'500
	70 % load	kg/h	17'600	22'300	27'900	35'100	38'500	44'600	52'700
	Temp. turbine outlet at 100 % load	°C	365	365	365	365	365	365	365
	Temp. turbine outlet at 85 & 70 % load	°C	355/350	360/360	360/360	355/350	365/365	360/360	355/350
4.0	Charge air : at 100 % load								
	Quantity at 45 °C & 60 % r.h.	kg/h	22'000	29'300	36'600	43'900	51'300	58'600	65'900
	Quantity at 27 °C & 60 % r.h.	kg/h	23'200	30'900	38'600	46'400	54'100	61'800	69'600
5.0	Starting air (max. pressure 30 bar)		Capacity of air bottles and air compressors on separate sheet						
6.0	Turning gear : El. mot. rat. 1500/1800 rev/min	kW	4/4.8	4/4.8	4/4.8	4/4.8	5.5/6.6	5.5/6.6	5.5/6.6
7.0	Engine radiation losses : at 100 % load 5)	kW	140	185	236	280	325	376	420
8.0	Heaviest engine part for :								
	Maintenance (turbocharger)	t	1.7	1.7	3	1.7	1.7	1.7	3
	Erection (engine dry, without flywheel)	t	50	65	73	81	93	104	119

Remarks : The figures given in the above table are subject to modifications. They apply to the requirement of the engine only
The need for FW generator has not been considered.
The values of heat dissipation are for determining the cooler sizes only, but not for heat balance purposes

1) Engine output at sea level, for ambient air up to 45 °C and 60 % relative humidity, sea water inlet up to 32 °C
2) The pumps are separately driven (engine driven pumps are an optional extra. For more details see overleaf)
3) Final pump head to be determined according to actual layout of piping installation
4) Max. admissible back pressure at turbine outlet 400 mm w.g.
5) Approximate values for engine only.

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Appendix: C.6.5

Roxana Bank Voyage Fuelling Returns

(litres)

	daily cons		total cons		running hrs	
voyage dates	hfo	mdo	hfo	mdo	hfo	mdo
10Dec88- 30Jan89	7355	3979	362565	196127	1150	24
02/03 - 11/04:89	5535	4847	217476	190463	843	91
19/04 - 22/05:89	5900	4723	192517	154102	681	81
27/05 - 11/07:89	6425	4618	285906	205501	839	212
19/07 - 28/08:89	7240	1955	291455	78710	890	63
05/09 - 08/10:89	7907	1074	264275	32220	756	32

(Source:Irvin and Johnson Voyage Fuelling Returns - Roxana Bank

Appendix: D.1.1

Steam Tables

p	t_s	v_g	h_f	h_{fg}	h_g	s_f	s_g
kPa	°C	m³/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg. K	kJ/kg. K
1,0	7,0	129,20	29	2 485	2 514	0,106	8,977
1,5	13,0	87,98	55	2 471	2 526	0,196	8,829
2,0	17,5	67,01	74	2 460	2 534	0,261	8,725
2,5	21,1	54,26	89	2 452	2 541	0,312	8,644
3,0	24,1	45,67	101	2 445	2 546	0,354	8,579
3,5	26,7	39,48	112	2 439	2 551	0,391	8,523
4,0	29,0	34,80	121	2 433	2 554	0,423	8,476
4,5	31,0	31,14	130	2 428	2 558	0,451	8,434
5,0	32,9	28,19	138	2 424	2 562	0,476	8,396
5,5	34,6	25,77	145	2 420	2 565	0,500	8,362
6,0	36,2	23,74	152	2 416	2 568	0,521	8,331
6,5	37,7	22,02	158	2 413	2 571	0,541	8,303
7,0	39,0	20,53	163	2 410	2 573	0,559	8,277
7,5	40,3	19,24	169	2 406	2 575	0,576	8,252
8,0	41,6	18,10	174	2 403	2 577	0,593	8,230
8,5	42,7	17,10	179	2 401	2 580	0,608	8,208
9,0	43,8	16,20	183	2 398	2 581	0,622	8,188
9,5	44,8	15,40	188	2 395	2 583	0,636	8,169
10	45,8	14,67	192	2 393	2 585	0,649	8,151
11	47,7	13,42	200	2 388	2 588	0,674	8,118
12	49,5	12,36	207	2 384	2 591	0,696	8,087
13	51,1	11,47	214	2 380	2 594	0,717	8,059
14	52,6	10,69	220	2 377	2 597	0,737	8,033
15	54,0	10,02	226	2 373	2 599	0,755	8,009
16	55,3	9,433	232	2 370	2 602	0,772	7,987
17	56,6	8,911	237	2 367	2 604	0,788	7,966
18	57,8	8,445	242	2 364	2 606	0,804	7,946
19	59,0	8,027	247	2 361	2 608	0,818	7,927
20	60,1	7,650	252	2 358	2 610	0,832	7,909
21	61,1	7,307	256	2 356	2 612	0,845	7,893
22	62,2	6,995	260	2 353	2 613	0,858	7,876
23	63,1	6,709	264	2 351	2 615	0,870	7,861
24	64,1	6,447	268	2 349	2 617	0,882	7,846
25	65,0	6,204	272	2 346	2 618	0,893	7,832
26	65,9	5,980	276	2 344	2 620	0,904	7,819
27	66,7	5,772	279	2 342	2 621	0,915	7,806
28	67,6	5,579	283	2 340	2 623	0,925	7,793
29	68,3	5,398	286	2 338	2 624	0,935	7,781
30	69,1	5,229	289	2 336	2 625	0,944	7,770
32	70,6	4,922	296	2 332	2 628	0,962	7,747
34	72,0	4,650	302	2 329	2 631	0,980	7,727
36	73,4	4,408	307	2 326	2 633	0,996	7,707
38	74,7	4,190	313	2 322	2 635	1,011	7,688
40	75,9	3,993	318	2 319	2 637	1,026	7,671
45	78,7	3,576	330	2 312	2 642	1,060	7,631
50	81,4	3,240	341	2 305	2 646	1,091	7,595
55	83,7	2,964	351	2 299	2 650	1,119	7,562
60	86,0	2,732	360	2 294	2 654	1,145	7,533
65	88,0	2,535	369	2 288	2 657	1,170	7,506
70	90,0	2,365	377	2 283	2 660	1,192	7,480

p	t_s	v_g	h_f	h_{fg}	h_g	s_f	s_g
kPa	°C	m³/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg. K	kJ/kg. K
75	91,8	2,217	385	2 279	2 664	1,213	7,457
80	93,5	2,087	392	2 274	2 666	1,233	7,435
85	95,2	1,972	399	2 270	2 669	1,252	7,415
90	96,7	1,869	405	2 266	2 671	1,270	7,395
95	98,2	1,777	412	2 262	2 674	1,287	7,377
100	99,6	1,694	418	2 258	2 676	1,303	7,360
110	102,3	1,549	429	2 251	2 680	1,333	7,328
120	104,8	1,428	439	2 244	2 683	1,361	7,300
130	107,1	1,325	449	2 238	2 687	1,387	7,272
140	109,3	1,236	458	2 232	2 690	1,411	7,247
150	111,4	1,159	467	2 226	2 693	1,434	7,223
160	113,3	1,091	475	2 221	2 696	1,455	7,202
170	115,2	1,031	483	2 216	2 699	1,475	7,181
180	116,9	0,977 2	491	2 211	2 702	1,494	7,162
190	118,6	0,929 0	498	2 206	2 704	1,513	7,144
200	120,2	0,885 4	505	2 202	2 707	1,530	7,127
210	121,8	0,845 9	511	2 197	2 708	1,547	7,111
220	123,3	0,809 8	518	2 193	2 711	1,563	7,095
230	124,7	0,776 8	524	2 189	2 713	1,578	7,080
240	126,1	0,746 5	530	2 185	2 715	1,593	7,066
250	127,4	0,718 4	535	2 181	2 716	1,607	7,052
260	128,7	0,692 5	541	2 177	2 718	1,621	7,039
270	130,0	0,668 4	546	2 174	2 720	1,634	7,026
280	131,2	0,646 0	551	2 170	2 721	1,647	7,014
290	132,4	0,625 1	557	2 167	2 724	1,660	7,002
300	133,5	0,605 6	561	2 163	2 724	1,672	6,991
310	134,7	0,587 2	566	2 160	2 726	1,683	6,980
320	135,8	0,570 0	571	2 157	2 728	1,695	6,969
330	136,8	0,553 8	576	2 154	2 730	1,706	6,959
340	137,9	0,538 5	580	2 150	2 730	1,717	6,949
350	138,9	0,524 0	584	2 147	2 731	1,727	6,939
360	139,9	0,510 3	589	2 144	2 733	1,738	6,930
370	140,8	0,497 4	593	2 141	2 734	1,748	6,921
380	141,8	0,485 1	597	2 139	2 736	1,757	6,912
390	142,7	0,473 4	601	2 136	2 737	1,767	6,903
400	143,6	0,462 2	605	2 133	2 738	1,776	6,894
410	144,5	0,451 6	609	2 130	2 739	1,786	6,886
420	145,4	0,441 5	612	2 128	2 740	1,795	6,878
430	146,3	0,431 8	616	2 125	2 741	1,803	6,870
440	147,1	0,422 6	620	2 122	2 742	1,812	6,862
450	147,9	0,413 8	624	2 120	2 744	1,820	6,855
460	148,7	0,405 3	627	2 117	2 744	1,829	6,847
470	149,5	0,397 2	630	2 115	2 745	1,837	6,840
480	150,3	0,389 4	634	2 112	2 746	1,845	6,833
490	151,1	0,381 9	637	2 110	2 747	1,853	6,826
500	151,8	0,374 7	640	2 107	2 747	1,860	6,819
520	153,3	0,361 1	647	2 103	2 750	1,875	6,806
540	154,8	0,348 5	653	2 098	2 751	1,890	6,793
560	156,2	0,336 7	659	2 094	2 753	1,904	6,781
580	157,5	0,325 7	665	2 089	2 754	1,918	6,769

p	t_s	v_g	h_f	h_{fg}	h_g	s_f	s_g
kPa	°C	m³/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg. K	kJ/kg. K
600	158,8	0,315 5	670	2 085	2 755	1,931	6,758
620	160,1	0,305 9	676	2 081	2 757	1,944	6,746
640	161,4	0,296 8	682	2 077	2 759	1,956	6,736
660	162,6	0,288 3	687	2 073	2 760	1,968	6,725
680	163,8	0,280 3	692	2 069	2 761	1,980	6,715
700	165,0	0,272 7	697	2 065	2 762	1,992	6,705
720	166,1	0,265 5	702	2 061	2 763	2,003	6,696
740	167,2	0,258 7	707	2 057	2 764	2,014	6,686
760	168,3	0,252 2	712	2 054	2 766	2,025	6,677
780	169,4	0,246 1	717	2 050	2 767	2,035	6,668
800	170,4	0,240 3	721	2 047	2 768	2,046	6,660
820	171,4	0,234 7	725	2 043	2 768	2,056	6,651
840	172,5	0,229 4	730	2 040	2 770	2,066	6,643
860	173,4	0,224 3	734	2 036	2 770	2,075	6,635
880	174,4	0,219 5	739	2 033	2 772	2,085	6,627
900	175,4	0,214 8	743	2 030	2 773	2,094	6,619
920	176,3	0,210 4	747	2 026	2 773	2,103	6,612
940	177,2	0,206 1	751	2 023	2 774	2,112	6,604
960	178,1	0,202 0	755	2 020	2 775	2,121	6,597
980	179,0	0,198 1	759	2 017	2 776	2,130	6,590
1 000	179,9	0,194 3	763	2 014	2 777	2,138	6,583
1 050	182,0	0,185 5	772	2 006	2 778	2,159	6,566
1 100	184,1	0,177 4	781	1 999	2 780	2,179	6,550
1 150	186,1	0,170 0	790	1 991	2 781	2,198	6,534
1 200	188,0	0,163 2	798	1 984	2 782	2,216	6,519
1 250	189,8	0,156 9	807	1 977	2 784	2,234	6,505
1 300	191,6	0,151 1	815	1 971	2 786	2,251	6,491
1 350	193,4	0,145 7	823	1 964	2 787	2,268	6,478
1 400	195,0	0,140 7	830	1 958	2 788	2,284	6,465
1 450	196,7	0,136 0	838	1 951	2 789	2,299	6,453
1 500	198,3	0,131 7	845	1 945	2 790	2,315	6,441
1 550	199,9	0,127 5	852	1 939	2 791	2,330	6,429
1 600	201,4	0,123 7	859	1 933	2 792	2,344	6,418
1 650	202,9	0,120 1	865	1 927	2 792	2,358	6,407
1 700	204,3	0,116 6	872	1 922	2 794	2,371	6,396
1 750	205,7	0,113 4	878	1 916	2 794	2,385	6,385
1 800	207,1	0,110 3	885	1 910	2 795	2,398	6,375
1 850	208,5	0,107 4	891	1 905	2 796	2,410	6,365
1 900	209,8	0,104 7	897	1 899	2 796	2,423	6,355
1 950	211,1	0,102 0	903	1 894	2 797	2,435	6,346
2 000	212,4	0,099 54	908	1 889	2 797	2,447	6,337
2 050	213,6	0,097 16	914	1 883	2 797	2,459	6,328
2 100	214,9	0,094 89	920	1 878	2 798	2,470	6,319
2 150	216,1	0,092 72	926	1 873	2 799	2,481	6,310
2 200	217,2	0,090 65	931	1 868	2 799	2,492	6,302
2 250	218,4	0,088 67	936	1 863	2 799	2,503	6,293
2 300	219,6	0,086 77	942	1 858	2 800	2,514	6,285
2 350	220,7	0,084 95	947	1 853	2 800	2,524	6,277
2 400	221,8	0,083 20	952	1 849	2 801	2,534	6,269
2 450	222,9	0,081 52	957	1 844	2 801	2,544	6,261

p	t_s	v_g	h_f	h_{fg}	h_g	s_f	s_g
kPa	°C	m³/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg. K	kJ/kg. K
2 500	223,9	0,079 91	962	1 839	2 801	2,554	6,254
2 550	225,0	0,078 35	967	1 834	2 801	2,564	6,246
2 600	226,0	0,076 86	972	1 830	2 802	2,574	6,239
2 650	227,1	0,075 41	977	1 825	2 802	2,583	6,232
2 700	228,1	0,074 02	981	1 821	2 802	2,592	6,224
2 750	229,1	0,072 68	986	1 816	2 802	2,602	6,217
2 800	230,1	0,071 39	991	1 811	2 802	2,611	6,210
2 850	231,0	0,070 14	995	1 807	2 802	2,620	6,203
2 900	232,0	0,068 93	1 000	1 802	2 802	2,628	6,197
2 950	232,9	0,067 76	1 004	1 798	2 802	2,637	6,190
3 000	233,8	0,066 63	1 008	1 794	2 802	2,646	6,184
3 100	235,7	0,064 47	1 017	1 785	2 802	2,662	6,171
3 200	237,5	0,062 44	1 025	1 777	2 802	2,679	6,159
3 300	239,2	0,060 53	1 034	1 768	2 802	2,695	6,146
3 400	240,9	0,058 73	1 042	1 760	2 802	2,710	6,134
3 500	242,5	0,057 03	1 050	1 752	2 802	2,725	6,123
3 600	244,2	0,055 41	1 058	1 744	2 802	2,740	6,112
3 700	245,8	0,053 89	1 065	1 736	2 801	2,755	6,100
3 800	247,3	0,052 44	1 073	1 728	2 801	2,769	6,090
3 900	248,8	0,051 06	1 080	1 721	2 801	2,783	6,079
4 000	250,3	0,049 75	1 087	1 713	2 800	2,797	6,069
4 100	251,8	0,048 50	1 095	1 705	2 800	2,810	6,058
4 200	253,2	0,047 31	1 102	1 698	2 800	2,823	6,048
4 300	254,7	0,046 17	1 109	1 690	2 799	2,836	6,038
4 400	256,1	0,045 08	1 115	1 683	2 798	2,849	6,029
4 500	257,4	0,044 04	1 122	1 676	2 798	2,861	6,019
4 600	258,8	0,043 04	1 129	1 668	2 797	2,874	6,010
4 700	260,1	0,042 08	1 135	1 661	2 796	2,886	6,000
4 800	261,4	0,041 16	1 142	1 654	2 796	2,897	5,991
4 900	262,7	0,040 28	1 148	1 647	2 795	2,909	5,982
5 000	263,9	0,039 43	1 155	1 640	2 795	2,921	5,974
5 100	265,2	0,038 61	1 161	1 633	2 794	2,932	5,965
5 200	266,4	0,037 82	1 167	1 626	2 793	2,943	5,956
5 300	267,6	0,037 07	1 173	1 619	2 792	2,954	5,948
5 400	268,8	0,036 33	1 179	1 612	2 791	2,965	5,939
5 500	269,9	0,035 63	1 185	1 605	2 790	2,976	5,931
5 600	271,1	0,034 95	1 191	1 598	2 789	2,986	5,923
5 700	272,2	0,034 29	1 197	1 591	2 788	2,997	5,915
5 800	273,4	0,033 65	1 202	1 585	2 787	3,007	5,907
5 900	274,5	0,033 03	1 208	1 578	2 786	3,017	5,899
6 000	275,6	0,032 44	1 214	1 571	2 785	3,027	5,891
6 200	277,7	0,031 30	1 225	1 558	2 783	3,047	5,875
6 400	279,8	0,030 23	1 236	1 545	2 781	3,066	5,860
6 600	281,8	0,029 22	1 247	1 532	2 779	3,085	5,845
6 800	283,8	0,028 27	1 257	1 519	2 776	3,104	5,831
7 000	285,8	0,027 37	1 267	1 506	2 773	3,122	5,816
7 200	287,7	0,026 52	1 278	1 493	2 771	3,140	5,802
7 400	289,6	0,025 72	1 288	1 481	2 769	3,157	5,788
7 600	291,4	0,024 95	1 298	1 468	2 766	3,174	5,774
7 800	293,2	0,024 22	1 307	1 455	2 762	3,191	5,761

Steam tables #3

p	t_s	v_g	h_f	h_{fg}	h_g	s_f	s_g
kPa	°C	m³/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg. K	kJ/kg. K
8 000	295,0	0,023 53	1 317	1 443	2 760	3,208	5,747
8 200	296,7	0,022 86	1 327	1 430	2 757	3,224	5,734
8 400	298,4	0,022 23	1 336	1 418	2 754	3,240	5,721
8 600	300,1	0,021 63	1 345	1 406	2 751	3,256	5,708
8 800	301,7	0,021 05	1 355	1 393	2 748	3,271	5,695
9 000	303,3	0,020 50	1 364	1 381	2 745	3,287	5,682
9 200	304,9	0,019 96	1 373	1 369	2 742	3,302	5,669
9 400	306,4	0,019 45	1 382	1 356	2 738	3,317	5,657
9 600	308,0	0,018 97	1 391	1 344	2 735	3,332	5,644
9 800	309,5	0,018 49	1 399	1 332	2 731	3,346	5,632
10 000	311,0	0,018 04	1 408	1 320	2 728	3,361	5,620
10 400	313,4	0,017 18	1 425	1 295	2 720	3,389	5,596
10 800	316,7	0,016 39	1 442	1 271	2 713	3,417	5,572
11 200	319,4	0,015 64	1 459	1 247	2 706	3,444	5,548
11 600	322,1	0,014 94	1 475	1 222	2 697	3,471	5,524
12 000	324,7	0,014 28	1 492	1 197	2 689	3,497	5,500
12 400	327,2	0,013 66	1 508	1 173	2 681	3,523	5,477
12 800	329,6	0,013 08	1 524	1 148	2 672	3,549	5,453
13 200	332,0	0,012 52	1 540	1 122	2 662	3,574	5,429
13 600	334,4	0,012 00	1 556	1 097	2 653	3,599	5,405
14 000	336,6	0,011 50	1 572	1 071	2 643	3,624	5,380
14 400	338,9	0,011 02	1 587	1 044	2 631	3,649	5,356
14 800	341,1	0,010 56	1 603	1 018	2 621	3,674	5,331
15 200	343,2	0,010 12	1 619	990	2 609	3,698	5,305
15 600	345,3	0,009 707	1 635	963	2 598	3,723	5,279
16 000	347,3	0,009 308	1 651	934	2 585	3,747	5,253
16 400	349,3	0,008 925	1 667	906	2 573	3,772	5,227
16 800	351,3	0,008 553	1 683	876	2 559	3,797	5,199
17 200	353,2	0,008 191	1 700	844	2 544	3,824	5,171
17 600	355,1	0,007 839	1 718	812	2 530	3,850	5,143
18 000	357,0	0,007 498	1 735	779	2 514	3,877	5,113
18 400	358,8	0,007 165	1 752	745	2 497	3,903	5,082
18 800	360,6	0,006 839	1 770	710	2 480	3,929	5,050
19 200	362,3	0,006 517	1 788	673	2 461	3,957	5,016
19 600	364,0	0,006 198	1 807	634	2 441	3,985	4,980
20 000	365,7	0,005 877	1 827	592	2 419	4,015	4,941
20 400	367,4	0,005 548	1 848	545	2 393	4,047	4,898
20 800	369,0	0,005 205	1 873	492	2 365	4,084	4,850
21 200	370,6	0,004 831	1 902	427	2 329	4,128	4,792
21 600	372,2	0,004 392	1 940	342	2 282	4,186	4,715
22 000	373,7	0,003 728	2 011	185	2 196	4,295	4,580
22 120	374,2	0,003 17	2 107	0	2 107	4,443	4,443

Appendix: D.1.2

Properties of Metals

Material	k, Btu/hr-ft-°F				c_p, Btu/lbm-°F	ρ, lbm/ft^3	α, ft^2/hr
	32 °F 0 °C	212 °F 100 °C	572 °F 300 °C	932 °F 500 °C	32 °F 0 °C	32 °F 0 °C	32 °F 0 °C
Metals—Pure							
Aluminum	117	119	133	156	0.208	169	3.33
Copper	224	218	212	207	0.091	558	4.42
Gold	169	170	...	...	0.030	1203	4.68
Iron	35.8	36.6	...	...	0.104	491	0.70
Lead	20.1	19	18	...	0.030	705	0.95
Magnesium	91	92	...	...	0.232	109	3.60
Molybdenum	72	68	64	62	0.060	638	1.88
Nickel	54	48	37	...	0.106	556	0.92
Silver	241	240	...	...	0.056	655	6.57
Tin	38	34	...	...	0.054	456	1.54
Zinc	65.1	63	58	...	0.091	446	1.60
Alloys							
Admiralty metal	65	64					
Brass, 70% Cu, 30% Zn	61.5	74	85	...	0.092	532	1.26
Bronze, 75% Cu, 25% Sn	15	...	...	...	0.082	541	0.34
Cast iron, Plain	33	31.8	27.7	24.8	0.11	474	0.63
Alloy	30	28.3	27	...	0.10	455	0.66
Constantan, 60% Cu, 40% Ni	12.4	12.8	...	...	0.10	557	0.22
18-8 stainless steel, Type 304	8.0	9.4	10.9	12.4	0.11	488	0.15
Type 347	8.0	9.3	11.0	12.8	0.11	488	0.15
Steel, mild, 1% C	26.5	26	25	22	0.11	490	0.49
SI Units	W/m·K				J/kg-K	kg/m^3	m^2/s
To convert to SI units multiply tabulated values by	1.729577				4.184 $\times 10^3$	1.601846 $\times 10^1$	2.580640 $\times 10^{-5}$

(Source: Pitts D.R. and Sissom L.E. 1977 Heat Transfer
McGraw Hill Book Company)

Appendix: D.1.3 #1

Properties of Nonmetals

Substance	T, °F	T, °C	c_p, Btu/lbm-°F	ρ, lbm/ft^3	k, Btu/hr-ft-°F	α, ft^2/hr
Structural						
Asphalt	68	20			0.43	
Bakelite	68	20	0.38	79.5	0.134	0.0044
Bricks						
Common	68	20	0.20	100	0.40	0.02
Face	68	20		128	0.76	
Carborundum brick	1110	600			10.7	
	2550	1400			6.4	
Chrome brick	392	200			1.34	0.036
	1022	550	0.20	188	1.43	0.038
	1652	900			1.15	0.031
Diatomaceous earth (fired)	400	205			0.14	
	1600	870			0.18	
Fireclay brick (burnt 2426 °F, 1330 °C)	932	500			0.60	0.020
	1472	800	0.23	128	0.62	0.021
	2012	1100			0.63	0.021
Fireclay brick (burnt 2642 °F, 1450 °C)	932	500			0.74	0.022
	1472	800	0.23	145	0.79	0.024
	2012	1100			0.81	0.024
Fireclay brick (Missouri)	392	200			0.58	0.015
	1112	600	0.23	165	0.85	0.022
	2552	1400			1.02	0.027
Magnesite	400	205			2.2	
	1200	650	0.27		1.6	
	2200	1205			1.1	
Cement, Portland				94	0.17	
Cement, mortar	75	24			0.67	
Concrete	68	20	0.21	119–144	0.47–0.81	0.019–0.027
Concrete, cinder	75	24			0.44	
Glass, plate	68	20	0.2	169	0.44	0.013
Glass, borosilicate	86	30		139	0.63	
Plaster, gypsum	70	21	0.2	90	0.28	0.016
Plaster, metal lath	70	21			0.27	
Plaster, wood lath	70	21			0.16	
Stone						
Granite			0.195	165	1.0–2.3	0.031–0.071
Limestone	210–570	100–300	0.217	155	0.73–0.77	0.022–0.023
Marble	68	20	0.193	156–169	1.6	0.054
Sandstone	68	20	0.17	135–144	0.94–1.2	0.041–0.049
SI Units			J/kg-K	kg/m^3	W/m-K	m^2/s
To convert to SI units multiply tabulated values by			4.184×10^3	1.601846×10^1	1.729577	2.580640×10^{-5}

Substance	T, °F	T, °C	c_p, Btu/lbm-°F	ρ, lbm/ft³	k, Btu/hr-ft-°F	α, ft²/hr
Structural (cont.)						
Wood, cross grain:						
Balsa	86	30		8.8	0.032	
Cypress	86	30		29	0.056	
Fir	75	24	0.65	26.0	0.063	0.0037
Oak	86	30	0.57	38–30	0.096	0.0049
Yellow pine	75	24	0.67	40	0.085	0.0032
White pine	86	30		27	0.065	
Wood, radial:						
Oak	68	20	0.57	38–30	0.10–0.12	0.0043–0.0047
Fir	68	20	0.65	26.0–26.3	0.08	0.0048
Insulating						
Asbestos	−328	−200		29.3	0.043	
	32	0			0.090	
Asbestos	32	0		36.0	0.087	
	212	100			0.111	
	392	200			0.120	
	752	400			0.129	
Asbestos	−328	−200		43.5	0.09	
	32	0			0.135	
Asbestos cement					1.2	
Asbestos sheet	124	51			0.096	
Asbestos felt (40 laminations per inch)	100	38			0.033	
	300	149			0.040	
	500	260			0.048	
Asbestos felt (20 laminations per inch)	100	38			0.045	
	300	149			0.055	
	500	260			0.065	
Balsam wool	90	32		2.2	0.023	
Cardboard, corrugated					0.037	
Celotex	90	32			0.028	
Corkboard	86	30		10	0.025	
Cork, ground	86	30		9.4	0.025	
SI Units			J/kg·K	kg/m³	W/m-K	m²/s
To convert to SI units multiply tabulated values by			4.184×10^3	1.601846×10^1	1.729577	2.580640×10^{-5}

Nonmetals #3

Substance	T, °F	T, °C	c_p, Btu/lbm-°F	ρ, lbm/ft^3	k, Btu/hr-ft-°F	α, ft^2/hr
Insulating (cont.)						
Diatomaceous earth (powdered)	200	93		14	0.033	
	400	204			0.039	
	600	316			0.046	
Felt, hair	20	−7		11.4	0.0212	
	100	38			0.0254	
	200	93			0.0299	
Fiber insulating board	70	21		14.8	0.028	
Glass wool	20	−7		1.5	0.0217	
	100	38			0.0313	
	200	93			0.0435	
Glass wool	20	−7		4.0	0.0179	
	100	38			0.0239	
	200	93			0.0317	
Glass wool	20	−7		6.0	0.0163	
	100	38			0.0218	
	200	93			0.0288	
Kapok	86	30			0.020	
Magnesia, 85%	100	38		16.9	0.039	
	200	93			0.041	
	300	149			0.043	
	400	204			0.046	
Rock wool	20	−7		4.0	0.0150	
	100	38			0.0224	
	200	93			0.0317	
Rock wool	20	−7		8.0	0.0171	
	100	38			0.0228	
	200	93			0.0299	
Rock wool	20	−7		12.0	0.0183	
	100	38			0.0226	
	200	93			0.0281	
Miscellaneous						
Aerogel, silica	248	120		8.5	0.013	
Clay	68	20	0.21	91.0	0.739	0.039
Coal, anthracite	68	20	0.30	75–94	0.15	0.005–0.006
Coal, powdered	86	30	0.31	46	0.067	0.005
Cotton	68	20	0.31	5	0.034	0.075
Earth, coarse	68	20	0.44	128	0.30	0.0054
Ice	32	0	0.46	57	1.28	0.048
Rubber, hard	32	0		74.8	0.087	
Sawdust	75	24			0.034	
Silk	68	20	0.33	3.6	0.021	0.017
SI Units			J/kg-K	kg/m^3	W/m-K	m^2/s
To convert to SI units multiply tabulated values by			4.184 $\times 10^3$	1.601846 $\times 10^1$	1.729577	2.580640 $\times 10^{-5}$

(Source: Pitts D.R. and Sissom L.E. 1977 Heat Transfer McGraw Hill Book Company)

Appendix: D.1.4 #1

Properties of Liquids in Saturated State

T, °F	T, °C	ρ, lbm/ft³	c_p, Btu/lbm-°F	ν, ft²/sec	k, Btu/hr-ft-°F	α, ft²/hr	Pr	β, 1/°R
				Water (H_2O)				
32	0	62.57	1.0074	1.925×10^{-5}	0.319	5.07×10^{-3}	13.6	
68	20	62.46	0.9988	1.083	0.345	5.54	7.02	0.10×10^{-3}
104	40	62.09	0.9980	0.708	0.363	5.86	4.34	
140	60	61.52	0.9994	0.514	0.376	6.02	3.02	
176	80	60.81	1.0023	0.392	0.386	6.34	2.22	
212	100	59.97	1.0070	0.316	0.393	6.51	1.74	
248	120	59.01	1.015	0.266	0.396	6.62	1.446	
284	140	57.95	1.023	0.230	0.395	6.68	1.241	
320	160	56.79	1.037	0.204	0.393	6.70	1.099	
356	180	55.50	1.055	0.186	0.390	6.68	1.004	
392	200	54.11	1.076	0.172	0.384	6.61	0.937	
428	220	52.59	1.101	0.161	0.377	6.51	0.891	
464	240	50.92	1.136	0.154	0.367	6.35	0.871	
500	260	49.06	1.182	0.148	0.353	6.11	0.874	
537	280	46.98	1.244	0.145	0.335	5.74	0.910	
572	300	44.59	1.368	0.145	0.312	5.13	1.019	
				Ammonia (NH_3)				
−58	−50	43.93	1.066	0.468×10^{-5}	0.316	6.75×10^{-3}	2.60	
−40	−40	43.18	1.067	0.437	0.316	6.88	2.28	
−22	−30	42.41	1.069	0.417	0.317	6.98	2.15	
−4	−20	41.62	1.077	0.410	0.316	7.05	2.09	
14	−10	40.80	1.090	0.407	0.314	7.07	2.07	
32	0	39.96	1.107	0.402	0.312	7.05	2.05	
50	10	39.09	1.126	0.396	0.307	6.98	2.04	
68	20	38.19	1.146	0.386	0.301	6.88	2.02	1.36×10^{-3}
86	30	37.23	1.168	0.376	0.293	6.75	2.01	
104	40	36.27	1.194	0.366	0.285	6.59	2.00	
122	50	35.23	1.222	0.355	0.275	6.41	1.99	
				Carbon dioxide (CO_2)				
−58	−50	72.19	0.44	0.128×10^{-5}	0.0494	1.558×10^{-3}	2.96	
−40	−40	69.78	0.45	0.127	0.0584	1.864	2.46	
−22	−30	67.22	0.47	0.126	0.0645	2.043	2.22	
−4	−20	64.45	0.49	0.124	0.0665	2.110	2.12	
14	−10	61.39	0.52	0.122	0.0635	1.989	2.20	
SI Units		kg/m³	J/kg-K	m²/s	W/m-K	m²/s	—	1/K
To convert to SI units multiply tabulated values by		1.601846×10^{1}	4.184×10^{3}	9.290304×10^{-2}	1.729577	2.580640×10^{-5}	—	1.80

T, °F	T, °C	ρ, lbm/ft³	c_p, Btu/lbm-°F	ν, ft²/sec	k, Btu/hr-ft-°F	α, ft²/hr	Pr	β, 1/°R
Carbon dioxide (CO_2) (cont.)								
32	0	57.87	0.59	0.117	0.0604	1.774	2.38	
50	10	53.69	0.75	0.109	0.0561	1.398	2.80	
68	20	48.23	1.2	0.098	0.0504	0.860	4.10	7.78×10^{-3}
86	30	37.32	8.7	0.086	0.0406	0.108	28.7	
Sulfur dioxide (SO_2)								
−58	−50	97.44	0.3247	0.521×10^{-5}	0.140	4.42×10^{-3}	4.24	
−40	−40	95.94	0.3250	0.456	0.136	4.38	3.74	
−22	−30	94.43	0.3252	0.399	0.133	4.33	3.31	
−4	−20	92.93	0.3254	0.349	0.130	4.29	2.93	
14	−10	91.37	0.3255	0.310	0.126	4.25	2.62	
32	0	89.80	0.3257	0.277	0.122	4.19	2.38	
50	10	88.18	0.3259	0.250	0.118	4.13	2.18	
68	20	86.55	0.3261	0.226	0.115	4.07	2.00	1.08×10^{-3}
86	30	84.86	0.3263	0.204	0.111	4.01	1.83	
104	40	82.98	0.3266	0.186	0.107	3.95	1.70	
122	50	81.10	0.3268	0.174	0.102	3.87	1.61	
Methylchloride (CH_3Cl)								
−58	−50	65.71	0.3525	0.344×10^{-5}	0.124	5.38×10^{-3}	2.31	
−40	−40	64.51	0.3541	0.342	0.121	5.30	2.32	
−22	−30	63.46	0.3564	0.338	0.117	5.18	2.35	
−4	−20	62.39	0.3593	0.333	0.113	5.04	2.38	
14	−10	61.27	0.3629	0.329	0.108	4.87	2.43	
32	0	60.08	0.3673	0.325	0.103	4.70	2.49	
50	10	58.83	0.3726	0.320	0.099	4.52	2.55	
68	20	57.64	0.3788	0.315	0.094	4.31	2.63	
86	30	56.38	0.3860	0.310	0.089	4.10	2.72	
104	40	55.13	0.3942	0.303	0.083	3.86	2.83	
122	50	53.76	0.4034	0.295	0.077	3.57	2.97	
Dichlorodifluoromethane (Freon = 12)(CCl_2F_2)								
−58	−50	96.56	0.2090	0.334×10^{-5}	0.039	1.94×10^{-3}	6.2	1.46×10^{-3}
−40	−40	94.81	0.2113	0.300	0.040	1.99	5.4	
−22	−30	92.99	0.2139	0.272	0.040	2.04	4.8	
−4	−20	91.18	0.2167	0.253	0.041	2.09	4.4	
14	−10	89.24	0.2198	0.238	0.042	2.13	4.0	
SI Units		kg/m³	J/kg-K	m²/s	W/m-K	m²/s	—	1/K
To convert to SI units multiply tabulated values by		1.601846×10^{1}	4.184×10^{3}	9.290304×10^{-2}	1.729577	2.580640×10^{-5}	—	1.80

T, °F	T, °C	ρ, $\frac{lbm}{ft^3}$	c_p, $\frac{Btu}{lbm\text{-}°F}$	ν, $\frac{ft^2}{sec}$	k, $\frac{Btu}{hr\text{-}ft\text{-}°F}$	α, $\frac{ft^2}{hr}$	Pr	β, $\frac{1}{°R}$
				Dichlorodifluoromethane (Freon = 12)(CCl_2F_2) (cont.)				
32	0	87.24	0.2232	0.230	0.042	2.16	3.8	
50	10	85.17	0.2268	0.219	0.042	2.17	3.6	
68	20	83.04	0.2307	0.213	0.042	2.17	3.5	
86	30	80.85	0.2349	0.209	0.041	2.17	3.5	
104	40	78.48	0.2393	0.206	0.040	2.15	3.5	
122	50	75.91	0.2440	0.204	0.039	2.11	3.5	
				Eutectic calcium chloride solution (29.9% $CaCl_2$)				
−58	−50	82.39	0.623	39.13×10^{-5}	0.232	4.52×10^{-3}	312	
−40	−40	82.09	0.6295	26.88	0.240	4.65	208	
−22	−30	81.79	0.6356	18.49	0.248	4.78	139	
−4	−20	81.50	0.642	11.88	0.257	4.91	87.1	
14	−10	81.20	0.648	7.49	0.265	5.04	53.6	
32	0	80.91	0.654	4.73	0.273	5.16	33.0	
50	10	80.62	0.660	3.61	0.280	5.28	24.6	
68	20	80.32	0.666	2.93	0.288	5.40	19.6	
86	30	80.03	0.672	2.44	0.295	5.50	16.0	
104	40	79.73	0.678	2.07	0.302	5.60	13.3	
122	50	79.44	0.685	1.78	0.309	5.69	11.3	
				Glycerin [$C_3H_5(OH)_3$]				
32	0	79.66	0.540	0.0895	0.163	3.81×10^{-3}	84.7×10^3	
50	10	79.29	0.554	0.0323	0.164	3.74	31.0	
68	20	78.91	0.570	0.0127	0.165	3.67	12.5	0.28×10^{-3}
86	30	78.54	0.584	0.0054	0.165	3.60	5.38	
104	40	78.16	0.600	0.0024	0.165	3.54	2.45	
122	50	77.72	0.617	0.0016	0.166	3.46	1.63	
				Ethylene glycol [$C_2H_4(OH_2)$]				
32	0	70.59	0.548	61.92×10^{-5}	0.140	3.62×10^{-3}	615	
68	20	69.71	0.569	20.64	0.144	3.64	204	0.36×10^{-3}
104	40	68.76	0.591	9.35	0.148	3.64	93	
140	60	67.90	0.612	5.11	0.150	3.61	51	
176	80	67.27	0.633	3.21	0.151	3.57	32.4	
212	100	66.08	0.655	2.18	0.152	3.52	22.4	
SI Units		$\frac{kg}{m^3}$	$\frac{J}{kg\text{-}K}$	$\frac{m^2}{s}$	$\frac{W}{m\text{-}K}$	$\frac{m^2}{s}$	—	$\frac{1}{K}$
To convert to SI units multiply tabulated values by		1.601846×10^1	4.184×10^3	9.290304×10^{-2}	1.729577	2.580640×10^{-5}	—	1.80

Saturated Liquids #4

T, °F	°C	ρ, lbm/ft³	c_p, Btu/lbm-°F	ν, ft²/sec	k, Btu/hr-ft-°F	α, ft²/hr	Pr	β, 1/°R
Engine oil (unused)								
32	0	56.13	0.429	0.0461	0.085	3.53×10^{-3}	47100	
68	20	55.45	0.449	0.0097	0.084	3.38	10400	0.39×10^{-3}
104	40	54.69	0.469	0.0026	0.083	3.23	2870	
140	60	53.94	0.489	0.903×10^{-3}	0.081	3.10	1050	
176	80	53.19	0.509	0.404	0.080	2.98	490	
212	100	52.44	0.530	0.219	0.079	2.86	276	
248	120	51.75	0.551	0.133	0.078	2.75	175	
284	140	51.00	0.572	0.086	0.077	2.66	116	
320	160	50.31	0.593	0.060	0.076	2.57	84	
Mercury (Hg)								
32	0	850.78	0.0335	0.133×10^{-5}	4.74	166.6×10^{-3}	0.0288	
68	20	847.71	0.0333	0.123	5.02	178.5	0.0249	1.01×10^{-4}
122	50	843.14	0.0331	0.112	5.43	194.6	0.0207	
212	100	835.57	0.0328	0.0999	6.07	221.5	0.0162	
302	150	828.06	0.0326	0.0918	6.64	246.2	0.0134	
392	200	820.61	0.0375	0.0863	7.13	267.7	0.0116	
482	250	813.16	0.0324	0.0823	7.55	287.0	0.0103	
600	316	802	0.032	0.0724	8.10	316	0.0083	
SI Units		kg/m³	J/kg-K	m²/s	W/m-K	m²/s	—	1/K
To convert to SI units multiply tabulated values by		1.601846×10^1	4.184×10^3	9.290304×10^{-2}	1.729577	2.580640×10^{-5}	—	1.80

(Source: Pitts D.R. and Sissom L.E. 1977 Heat Transfer McGraw Hill Book Company)

Appendix: D.1.5 #1

Properties of Gases at Atmospheric Pressure

T, °F	T, °C	ρ, $\frac{lbm}{ft^3}$	c_p, $\frac{Btu}{lbm\text{-}°F}$	μ_m, $\frac{lbm}{ft\text{-}sec}$	ν, $\frac{ft^2}{sec}$	k, $\frac{Btu}{hr\text{-}ft\text{-}°F}$	α, $\frac{ft^2}{hr}$	Pr
Air								
−280	−173	0.2248	0.2452	0.4653×10^{-5}	2.070×10^{-5}	0.005342	0.09691	0.770
−190	−123	0.1478	0.2412	0.6910	4.675	0.007936	0.2226	0.753
−100	−73	0.1104	0.2403	0.8930	8.062	0.01045	0.3939	0.739
−10	−23	0.0882	0.2401	1.074	10.22	0.01287	0.5100	0.722
80	27	0.0735	0.2402	1.241	16.88	0.01516	0.8587	0.708
170	77	0.0623	0.2410	1.394	22.38	0.01735	1.156	0.697
260	127	0.0551	0.2422	1.536	27.88	0.01944	1.457	0.689
350	177	0.0489	0.2438	1.669	31.06	0.02142	1.636	0.683
440	227	0.0440	0.2459	1.795	40.80	0.02333	2.156	0.680
530	277	0.0401	0.2482	1.914	47.73	0.02519	2.531	0.680
620	327	0.0367	0.2520	2.028	55.26	0.02692	2.911	0.680
710	377	0.0339	0.2540	2.135	62.98	0.02862	3.324	0.682
800	427	0.0314	0.2568	2.239	71.31	0.03022	3.748	0.684
890	477	0.0294	0.2593	2.339	79.56	0.03183	4.175	0.686
980	527	0.0275	0.2622	2.436	88.58	0.03339	4.631	0.689
1070	577	0.0259	0.2650	2.530	97.68	0.03483	5.075	0.692
1160	627	0.0245	0.2678	2.620	106.9	0.03628	5.530	0.696
1250	677	0.0232	0.2704	2.703	116.5	0.03770	6.010	0.699
1340	727	0.0220	0.2727	2.790	126.8	0.03901	6.502	0.702
1520	827	0.0200	0.2772	2.955	147.8	0.04178	7.536	0.706
1700	927	0.0184	0.2815	3.109	169.0	0.04410	8.514	0.714
1880	1027	0.0169	0.2860	3.258	192.8	0.04641	9.602	0.722
2060	1127	0.0157	0.2900	3.398	216.4	0.04880	10.72	0.726
2240	1227	0.0147	0.2939	3.533	240.3	0.05098	11.80	0.734
2420	1327	0.0138	0.2982	3.668	265.8	0.05348	12.88	0.741
2600	1427	0.0130	0.3028	3.792	291.7	0.05550	14.00	0.749
2780	1527	0.0123	0.3075	3.915	318.3	0.05750	15.09	0.759
2960	1627	0.0116	0.3128	4.029	347.1	0.0591	16.40	0.767
3140	1727	0.0110	0.3196	4.168	378.8	0.0612	17.41	0.783
3320	1827	0.0105	0.3278	4.301	409.9	0.0632	18.36	0.803
3500	1927	0.0100	0.3390	4.398	439.8	0.0646	19.05	0.831
3680	2027	0.0096	0.3541	4.513	470.1	0.0663	19.61	0.863
3860	2127	0.0091	0.3759	4.611	506.9	0.0681	19.92	0.916
4160	2293	0.0087	0.4031	4.750	546.0	0.0709	20.21	0.972
Helium								
−456	−271		1.242	5.66×10^{-7}		0.0061		
−400	−240	0.0915	1.242	33.7	3.68×10^{-5}	0.0204	0.1792	0.74
−200	−129	0.211	1.242	84.3	39.95	0.0536	2.044	0.70
−100	−73	0.0152	1.242	105.2	69.30	0.0680	3.599	0.694
0	−18	0.0119	1.242	122.1	102.8	0.0784	5.299	0.70
200	93	0.00829	1.242	154.9	186.9	0.0977	9.490	0.71
SI Units		$\frac{kg}{m^3}$	$\frac{J}{kg\text{-}K}$	$\frac{kg}{m\text{-}s}$	$\frac{m^2}{s}$	$\frac{W}{m\text{-}K}$	$\frac{m^2}{s}$	—
To convert to SI units multiply tabulated values by		1.601846×10^1	4.184×10^3	1.488164	9.290304×10^{-2}	1.729577	2.580640×10^{-5}	—

Gas Properties #2

T, °F	T, °C	ρ, lbm/ft³	c_p, Btu/lbm-°F	μ_m, lbm/ft-sec	ν, ft²/sec	k, Btu/hr-ft-°F	α, ft²/hr	Pr
Helium								
400	204	0.00637	1.242	184.8	289.9	0.114	14.40	0.72
600	316	0.00517	1.242	209.2	404.5	0.130	20.21	0.72
800	427	0.00439	1.242	233.5	531.9	0.145	25.81	0.72
1000	538	0.00376	1.242	256.5	682.5	0.159	34.00	0.72
1200	649	0.00330	1.242	277.9	841.0	0.172	41.98	0.72
Hydrogen								
−406	−243	0.05289	2.589	1.079×10^{-6}	2.040×10^{-5}	0.0132	0.0966	0.759
−370	−223	0.03181	2.508	1.691	5.253	0.0209	0.262	0.721
−280	−173	0.01534	2.682	2.830	18.45	0.0384	0.933	0.712
−190	−123	0.01022	3.010	3.760	36.79	0.0567	1.84	0.718
−100	−73	0.00766	3.234	4.578	59.77	0.0741	2.99	0.719
−10	−23	0.00613	3.358	5.321	86.80	0.0902	4.38	0.713
80	27	0.00511	3.419	6.023	117.9	0.105	6.02	0.706
170	77	0.00438	3.448	6.689	152.7	0.119	7.87	0.697
260	127	0.00383	3.461	7.300	190.6	0.132	9.95	0.690
350	177	0.00341	3.463	7.915	232.1	0.145	12.26	0.682
440	227	0.00307	3.465	8.491	276.6	0.157	14.79	0.675
530	277	0.00279	3.471	9.055	324.6	0.169	17.50	0.668
620	327	0.00255	3.472	9.599	376.4	0.182	20.56	0.664
800	427	0.00218	3.481	10.68	489.9	0.203	26.75	0.659
980	527	0.00191	3.505	11.69	612	0.222	33.18	0.664
1160	627	0.00170	3.540	12.62	743	0.238	39.59	0.676
1340	727	0.00153	3.575	13.55	885	0.254	46.49	0.686
1520	827	0.00139	3.622	14.42	1039	0.268	53.19	0.703
1700	927	0.00128	3.670	15.29	1192	0.282	60.00	0.715
1880	1027	0.00118	3.720	16.18	1370	0.296	67.40	0.733
1940	1060	0.00115	3.735	16.42	1429	0.300	69.80	0.736
Oxygen								
−280	−173	0.2492	0.2264	5.220×10^{-6}	2.095×10^{-5}	0.00522	0.09252	0.815
−190	−123	0.1635	0.2192	7.721	4.722	0.00790	0.2204	0.773
−100	−73	0.1221	0.2181	9.979	8.173	0.01054	0.3958	0.745
−10	−23	0.0975	0.2187	12.01	12.32	0.01305	0.6120	0.725
80	27	0.0812	0.2198	13.86	17.07	0.01546	0.8662	0.709
170	77	0.0695	0.2219	15.56	22.39	0.01774	1.150	0.702
260	127	0.0609	0.2250	17.16	28.18	0.02000	1.460	0.695
350	177	0.0542	0.2285	18.66	34.43	0.02212	1.786	0.694
440	227	0.0487	0.2322	20.10	41.27	0.02411	2.132	0.697
530	277	0.0443	0.2360	21.48	48.49	0.02610	2.496	0.700
620	327	0.0406	0.2399	22.79	56.13	0.02792	2.867	0.704
SI Units		kg/m³	J/kg-K	kg/m-s	m²/s	W/m-K	m²/s	—
To convert to SI units multiply tabulated values by		1.601846×10^{1}	4.184×10^{3}	1.488164	9.290304×10^{-2}	1.729577	2.580640×10^{-5}	—

T, °F	T, °C	ρ, $\frac{\text{lbm}}{\text{ft}^3}$	c_p, $\frac{\text{Btu}}{\text{lbm-°F}}$	μ_m, $\frac{\text{lbm}}{\text{ft-sec}}$	ν, $\frac{\text{ft}^2}{\text{sec}}$	k, $\frac{\text{Btu}}{\text{hr-ft-°F}}$	α, $\frac{\text{ft}^2}{\text{hr}}$	Pr
Nitrogen								
−280	−173	0.2173	0.2561	4.611×10^{-6}	2.122×10^{-5}	0.005460	0.09811	0.786
−100	−73	0.1068	0.2491	8.700	8.146	0.01054	0.3962	0.747
80	27	0.0713	0.2486	11.99	16.82	0.01514	0.8542	0.713
260	127	0.0533	0.2498	14.77	27.71	0.01927	1.447	0.691
440	227	0.0426	0.2521	17.27	40.54	0.02302	2.143	0.684
620	327	0.0355	0.2569	19.56	55.10	0.02646	2.901	0.686
800	427	0.0308	0.2620	21.59	70.10	0.02960	3.668	0.691
980	527	0.0267	0.2681	23.41	87.68	0.03241	4.528	0.700
1160	627	0.0237	0.2738	25.19	98.02	0.03507	5.404	0.711
1340	727	0.0213	0.2789	26.88	126.2	0.03741	6.297	0.724
1520	827	0.0194	0.2832	28.41	146.4	0.03958	7.204	0.736
1700	927	0.0178	0.2875	29.90	168.0	0.04151	8.111	0.748
Carbon dioxide								
−64	−53	0.1544	0.187	7.462×10^{-6}	4.833×10^{-5}	0.006243	0.2294	0.818
−10	−23	0.1352	0.192	8.460	6.257	0.007444	0.2868	0.793
80	27	0.1122	0.208	10.051	8.957	0.009575	0.4103	0.770
170	77	0.0959	0.215	11.561	12.05	0.01183	0.5738	0.755
260	127	0.0838	0.225	12.98	15.49	0.01422	0.7542	0.738
350	177	0.0744	0.234	14.34	19.27	0.01674	0.9615	0.721
440	227	0.0670	0.242	15.63	23.[illegible]	0.01937	1.195	0.702
530	277	0.0608	0.250	16.85	27.71	0.02208	1.453	0.685
620	327	0.0558	0.257	18.03	32.31	0.02491	1.737	0.668
Carbon monoxide								
−64	−53	0.09699	0.2491	9.295×10^{-6}	9.583×10^{-5}	0.01101	0.4557	0.758
−10	−23	0.0525	0.2490	10.35	12.14	0.01239	0.5837	0.750
80	27	0.07109	0.2489	11.990	16.87	0.01459	0.8246	0.737
170	77	0.06082	0.2492	13.50	22.20	0.01666	1.099	0.728
260	127	0.05329	0.2504	14.91	27.98	0.01864	1.397	0.722
350	177	0.04735	0.2520	16.25	34.32	0.0252	1.720	0.718
440	227	0.04259	0.2540	17.51	41.11	0.02232	2.063	0.718
530	277	0.03872	0.2569	18.74	48.40	0.02405	2.418	0.721
620	327	0.03549	0.2598	19.89	56.04	0.02569	2.786	0.724
Ammonia (NH_3)								
−58	−50	0.0239	0.525	4.875×10^{-6}	2.04×10^{-4}	0.0099	0.796	0.93
32	0	0.0495	0.520	6.285	1.27	0.0127	0.507	0.90
122	50	0.0405	0.520	7.415	1.83	0.0156	0.744	0.88
212	100	0.0349	0.534	8.659	2.48	0.0189	1.015	0.87
302	150	0.0308	0.553	9.859	3.20	0.0226	1.330	0.87
392	200	0.0275	0.572	11.08	4.03	0.0270	1.713	0.84
SI Units		$\frac{\text{kg}}{\text{m}^3}$	$\frac{\text{J}}{\text{kg-K}}$	$\frac{\text{kg}}{\text{m-s}}$	$\frac{\text{m}^2}{\text{s}}$	$\frac{\text{W}}{\text{m-K}}$	$\frac{\text{m}^2}{\text{s}}$	—
To convert to SI units multiply tabulated values by		1.601846×10^1	4.184×10^3	1.488164	9.290304×10^{-2}	1.729577	2.580640×10^{-5}	—

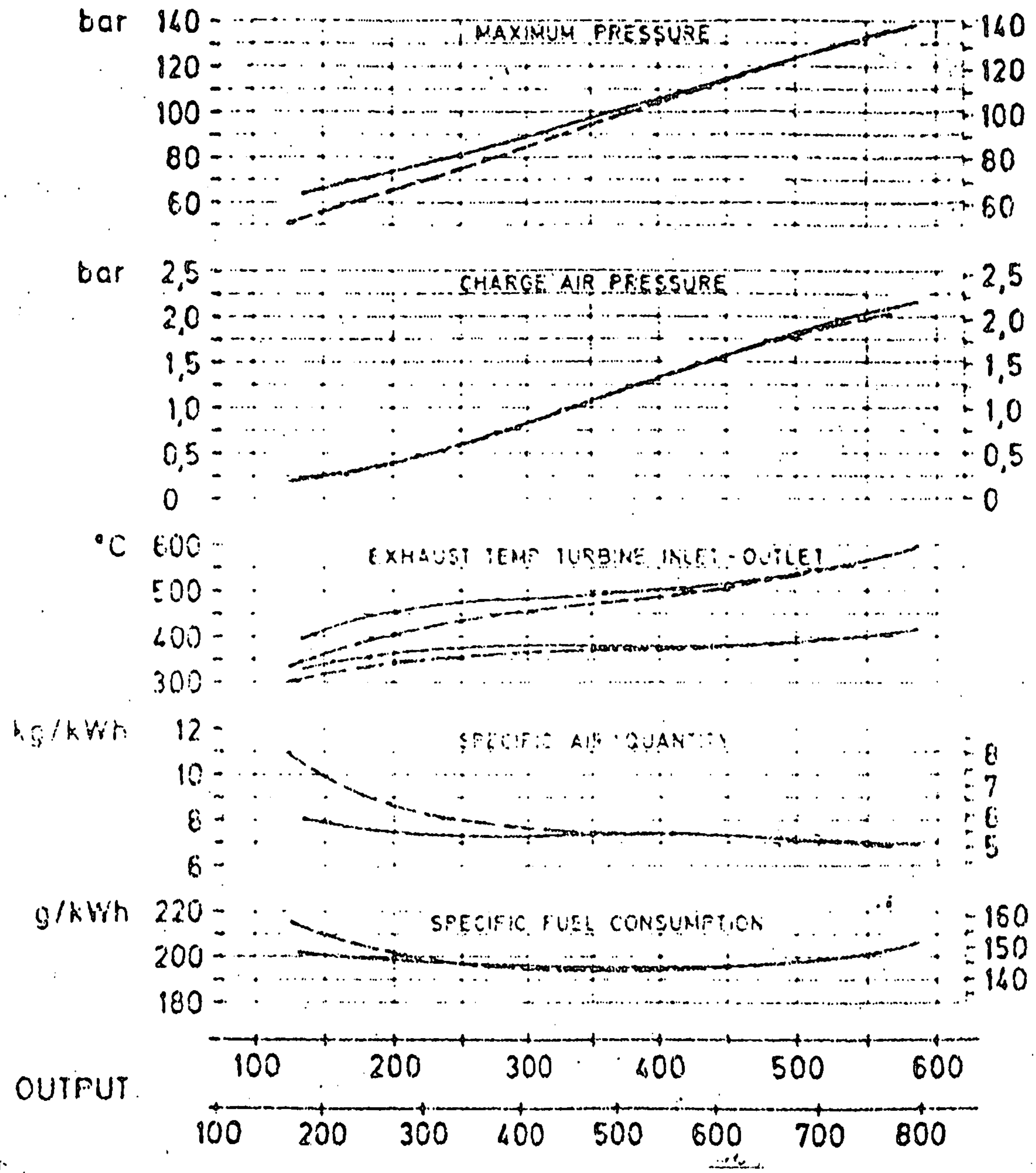
Gas Properties #4

T, °F	T, °C	ρ, $\frac{lbm}{ft^3}$	c_p, $\frac{Btu}{lbm\text{-}°F}$	μ_m, $\frac{lbm}{ft\text{-}sec}$	ν, $\frac{ft^2}{sec}$	k, $\frac{Btu}{hr\text{-}ft\text{-}°F}$	α, $\frac{ft^2}{hr}$	Pr
Steam (H_2O vapor)								
224	107	0.0366	0.492	8.54×10^{-6}	2.33×10^{-4}	0.0142	0.789	1.060
260	127	0.0346	0.481	9.03	2.61	0.0151	0.906	1.040
350	177	0.0306	0.473	10.25	3.35	0.0173	1.19	1.010
440	227	0.0275	0.474	11.45	4.16	0.0196	1.50	0.996
530	277	0.0250	0.477	12.66	5.06	0.0219	1.84	0.991
620	327	0.0228	0.484	13.89	6.09	0.0244	2.22	0.986
710	377	0.0211	0.491	15.10	7.15	0.0268	2.58	0.995
800	427	0.0196	0.498	16.30	8.31	0.0292	2.99	1.000
890	477	0.0183	0.506	17.50	9.56	0.0317	3.42	1.005
980	527	0.0171	0.514	18.72	10.98	0.0342	3.88	1.010
1070	577	0.0161	0.522	19.95	12.40	0.0368	4.38	1.019
SI Units		$\frac{kg}{m^3}$	$\frac{J}{kg\text{-}K}$	$\frac{kg}{m\text{-}s}$	$\frac{m^2}{s}$	$\frac{W}{m\text{-}K}$	$\frac{m^2}{s}$	—
To convert to SI units multiply tabulated values by		1.601846×10^1	4.184×10^3	1.488164	9.290304×10^{-2}	1.729577	2.580640×10^{-5}	—

(Source: Pitts D.R. and Sissom L.E. 1977 Heat Transfer McGraw Hill Book Company)

Appendix: D.1.6

Performance Data of the Sulzer Type Z 40/48 Engine



(Source: The Motorship 1978, A Special Survey:61)

Appendix E

Correspondence

REF. NO.: —

DATE: 29 - 09 - 93

Departement van Vervoer • Department of Transport

SHIRLEY CHAMBERS, BAKERSTRAAT/STREET, PRIVAATSAK/PRIVATE BAG X54309, DURBAN, 4000. TEL: 3071501 AD/ADD: "NAUTA" TELEKS/TELEX 620269, FAX: 3064983

TO / AAN: MANFRED K BERSZINSKI (ALFA-LAVAL)

ATTENTION / AANDAG:

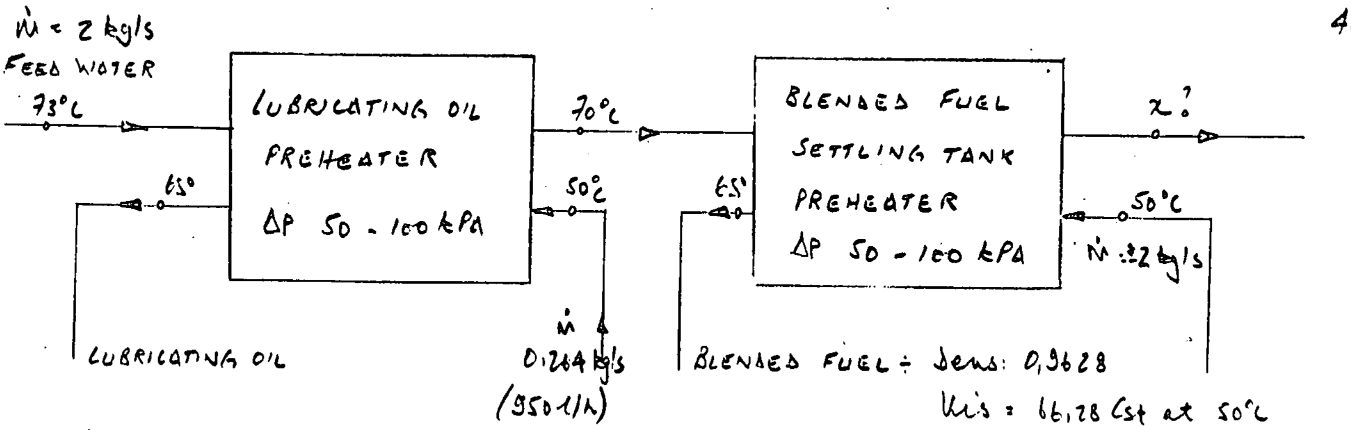
FAX NO OF RECEIVER / FAX NO VAN ONTVANGER: 974 - 7531

NAME OF SENDER / NAAM VAN AFSENDER: DAVID FIDDLER

OFFICE & EXTENSION NO. / KANTOOR-EN BYLYNNOMMER: 3071501 (03i)

THIS MEMO CONSISTS OF / HIERDIE MEMO BESTAAN UIT 1 PAGE(S) IN TOTAL, INCLUDING THIS PAGE. / BLADSY(E), INSLUITEND HIERDIE BLADSY.

MESSAGE: / BOODSKAP: As discussed in our telecom today, I am trying to reduce this feed water temperature to acceptable limits. A fresh water distiller preheater will be installed after the blended fuel heater to reduce the feed water to 40°C



I think it is safe to assume pressure drops of 50 – 100 kPa. In my opinion it will be better to circulate the blended fuel through a heat exchanger than to use large tank heating coils. Blended fuel should be circulated to ensure mixing.

Many thanks. David Fiddler

REF. NO.: DMF FAX 02/89

DATE: 09 - 10 - 89

Departement van Vervoer • Department of Transport

SHIRLEY CHAMBERS, BAKERSTRAAT/STREET, PRIVAATSAK/PRIVATE BAG X54309, DURBAN, 4000. TEL: 3071501 AD/ADD: "NAUTA" TELEKS/TELEX 620269, FAX: 3064983

TO
AAN: ALSA ENGINEERING (PTY) LTD

ATTENTION
AANDAG: MANFRED K BERSZINSKI

FAX NO OF RECEIVER
FAX NO VAN ONTVANGER: 974 - 7531

NAME OF SENDER
NAAM VAN AFSENDER: DAVID FIDDLER

OFFICE & EXTENSION NO.
KANTOOR-EN BYLYNNOMMER: (031) 3071501

THIS MEMO CONSISTS OF 2 PAGE(S) IN TOTAL, INCLUDING THIS PAGE.
HIERDIE MEMO BESTAAN UIT 2 BLADSY(E), INSLUITEND HIERDIE BLADSY.

MESSAGE:
BOODSKAP: I would be most grateful if you could give me specifications for a central cooler to handle the following water/water operations.

Max. heat load to be dissipated 2310 kW
Min. heat load to be dissipated 1490 kW.

Primary fluid - engine raw water inlet (max) 53,3°C out 32°C
inlet (min) 47,2°C out 32°C

Secondary fluid - sea water inlet (max) 30°C out ?
inlet (min) 16°C out ?

Primary fluid volume flow (max) 94,42 m^3/h
(min) 81,42 m^3/h

Secondary fluid volume flow (max) 130 m^3/h
(min) 89 m^3/h

FROM: DAVID FIDDLER (031) TEL 3021501 FAX 3064953

TO: M.K. BERSZINSKI

Pressure drops as close to 50 kPa as possible.

I have selected a pump system that will give maximum volume flow of 130 m³/h to handle maximum heat load at maximum seawater temperature. The pump will be two speed to give a volume flow of 89 m³/h which should be able to handle the minimum heat load and minimum. Pump combinations between maximum and minimum points will vary as sea temperature and heat load varies.

When doing an approximate calculation (Heat Exchanger Guide), I am unsure of the surface area obtained and hence number of plates to use, including the type of PHE. Your assistance will be appreciated.

David Fiddler.

P.O. Box 38027
POINT^
4069

20 October 1989

YOUR REF: EB 0229

Mr Colin Kerr
Engloserve (Pty) Ltd
P.O. Box 911
RANDBURG
2125

Dear Sir

THERMOPANELS

Thank you for your letter and the interest shown in my project.

To clarify the following points:

1. The curved tank side is in contact with the sea water, there is no insulation and the sea temperature is $\pm$ 15°C.

2. The fuel in the bunker tanks will be at $\pm$ 15°C as there is no tank heating. The settling tank is topped up once a day (although with fuel oil at 15°C I would imagine this would take some time) with 7.75 m^3 of H.F.O.

3. I had thought of pumping the 7.75 m^3 H.F.O. (at $\pm$ 15°C) into the settling tank at (45°C) which would reduce the settling tank temperature and raising the temperature from the resultant temperature to 45°, with the settling tank volume reducing by 7.75 m^3 over the 24 hours.

As my completed dissertation must contain acknowledgements do you have any objection to me acknowledging your Company? I am most grateful for your input.

Yours faithfully

D M Fiddler.

D.M. FIDDLER

☒ 4-22159 S.A.
Fax: (011) 787-5993

ENGCONSERVE (Pty) Ltd.
HEAT EXCHANGE PANELS • MADE IN SOUTH AFRICA

☎ (011) 787-5929

P. O. Box 911
Randburg 2125
Transvaal R.S.A.

Reg. No. 81/08829/07

Randburg Centre:
166 Hendrik Verwoerd Dr.
Randburg

Your Ref:

Our Ref: EB 0229

Mr. D.M. Fiddler,
P.O. Box 38027,
POINT
4069

Dear Mr. Fiddler,

'THERMOPANELS'

Thank you for your letter dated 29th. August 1989.

'Thermopanels' are suitable for your application and have many advantages over other types of heat exchangers.

Would you please clarify the following points so that we can calculate the most economic design.

1. Is the curved outer side (Ships side) of the H.F.O. settling tank in direct contact with the sea i.e. no insulation between H.F.O. and ship's side. If so, can we assume sea water temperature to be ± 5°C.

2. Can we assume that the settling tank is "topped up" with 7,75M³ of H.F.O. at ± 5°C from the bunker tanks in 24 hours.

3. We see the problem as initially raising 24,4M³ of H.F.O. in the settling tank from 37°C to 45°C and holding 16,65M³ (24,4M³ - 7,75M³) at 45°C. Then heating 7,75M³ from the bunker tanks from 5°C to 45°C daily.

Please let us know if we have interpreted your requirements correctly.

Yours faithfully,
ENGCONSERVE (PTY) LTD.

C. Kerr

COLIN KERR
DIRECTOR.

P.O.Box 38027
POINT. 4069
29 August 1989.

EngConserve (PTY) Ltd
P.O.Box 911
RANDBURG 2125.

Att: Mr.Colin Kerr.

Dear Sir,

1. I am a Marine Engineer and a Ship Surveyor (Engineer) by profession and am at present studying for a Masters Diploma in Mechanical Engineering at Natal Technikon using the research option.

2. My subject is "Energy Saving in Large Fishing Trawlers" where the vessels remain at sea for up to 50 days. In order to in- overall plant efficiency, I am investigating ways to recover some waste heat from the Main Engine cooling water and exhaust gas.

3. The vessels under investigation use blended fuel to run their Main Engines and consumption is around 7,75m3 per 24 hours. The fuel is pumped from the bunker tanks to a settling tank which has a maximum capacity of 24,4 m3. The usual practice is for the engineers to pump the tank once a day to 23 m3. A fuel oil purifier draws the fuel from the settling tank and discharges into a daily service tank which supplies the Main Engine fuel rail.

4. At present there is no heating in the settling tank and the purifier heater, service tank heater and inline fuel heater use electric elements - totalling 120 kW.
It is my intention to use circulating water to provide some heat to the settling tank to promote better seperation of water etc. There are two options for the supply of heated water:

4.1 From the lubricating oil cooler and charge air cooler. This cooling water is in the region of 48 degC and may be cooled to 32 degC. The volume fllow is 80 m3/hr (or a percentage of this) and the operating pressure is 300 kPa.
The fuel oil ideally should be heated from 37 to 45degC.

4.2 From hot feed water which has been heated by the engine jacket cooling water. This feed water temperature is in region of 73 degC and should be cooled to 40 degC. The volume flow is 7,5 m3/hr and operating pressure is 600 kPa

--- The physical dimensions of the settling tank are shown in the enclosed sketch.
The fuel particulars are: SG : 0,9628 at 20 degC
Viscosity: 666,28 Cst at 50 degC
The circulating water in both cases are fresh water.

5. My queries are:

5.1 Are your thermopanels suitable for the type of application that I envisage,
5.2 Would I obtain the temperature rises in the fuel oil required in 4.1, 4.2 over a period of 24 hours?

6. My intention is that to ensure that all modifications and heat recovery devices and equipment can be manufactured in South Africa by South African companies.
I intend submitting my dissertation for evaluation in January 1990.

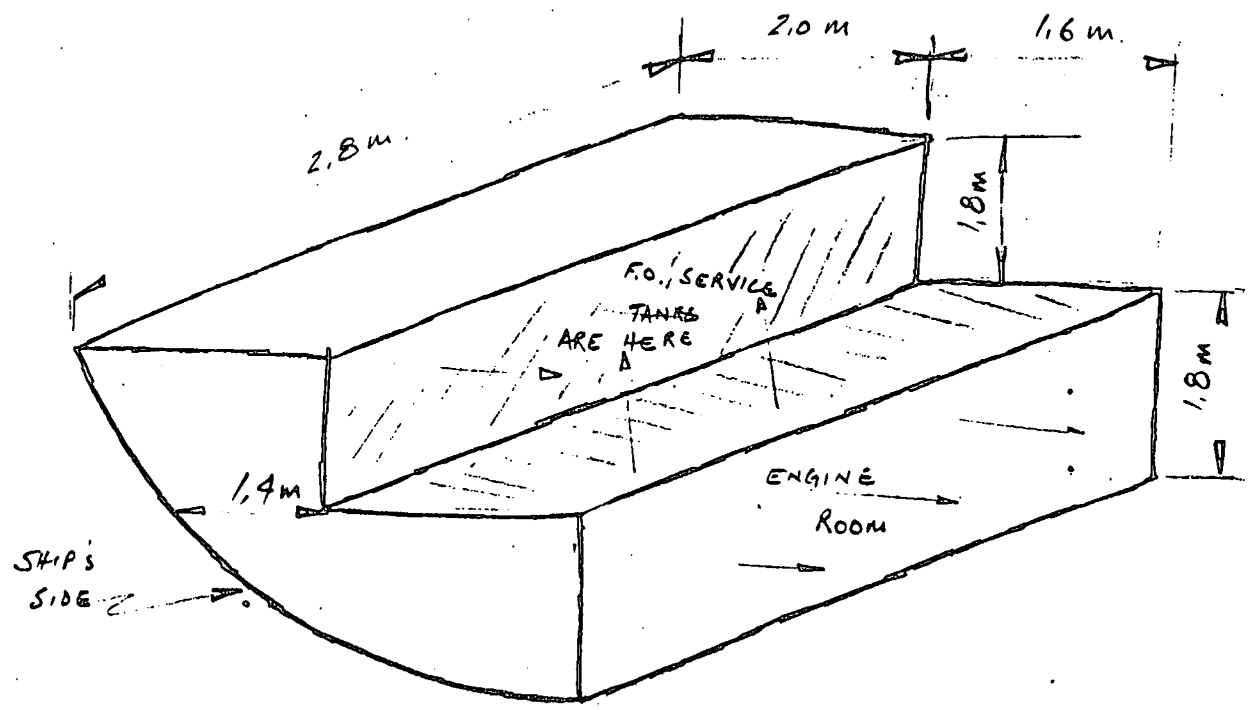
7. I wouuld be most gratefull if you could supply me with information on your Thermopanels as I would like to evaluate this system of heating against a circulating system using a heat exchanger and pumping system.

Yours faithfully,

DM Fiddler.

D.M.FIDDLER.

H.F.O. SETTLING TANK



TOTAL CAPACITY 24,4 m^3

{Measurements accurate within ± 0,1 m}

TANK ENDS ARE IN ENGINE ROOM.

UPPER TANK TOP FORMS PART OF THE UPPER DECK

P O BOX 38027
POINT (NATAL)
4069.
26 APRIL 1989

Mr. M. Berszinski
ALPHA - LAVAL
100 Electron Avenue
<u>ISANDO.</u>

Dear Sir,

I am a Marine Engineer employed by the Department of Transport (Marine Division) as a Ship Surveyor (Engineer). At present I am conducting a research project into energy saving in large freezer trawlers in order to obtain a Masters Diploma in Mechanical Engineering at Technikon Natal.

As my research is based on heat recovery, exclusive use of heat exchangers is envisaged. I have had experience with your heat exchangers and found them to be highly efficient and simple to maintain which will suit trawler operators.

I would be most gratefull if you could supply me with information on your heat exchangers for volume flows of 84m /h, 87m /h and 30m /h. My design is still in its infant stage as I am trying to match volume flows to the various components such as charge air coolers, jacket and lubricating oil coolers, etc.

A factor has arisen that in one particular heat exchanger the coolant volume flow is approximately 320m /h and the feed water volume flow is approximately 87m /h. Is this too higher a differential between the two liquids? I would also appreciate data on temperature gradients through the heat exchanger, fouling factors (if any) and heat transfer coefficients of the materials used in construction.

My final design will replace all shell and tube heat exchangers where applicable to improve efficiency of the heat transfer.

As I also intend investigating heat recovery from the large freezer plants fitted, it would be of great assistance to me if you could furnish me with the following information:

1) Are ALPHA LAVAL heat exchangers suitable for use as gas condensers after the refrigaration compressors and if so what are the limitations?

2) Is there an appreciable pressure drop across ALPHA LAVAL heat exchangers?

Your assistance will be greatly appreciated.

Yours faithfully,

D.M. FIDDLER.

WÄRTSILÄ DIESEL

Diesel Technology

K.Jofs/MMR 11 May 1989

D.M.Fiddler
P.O.Box 38027
Point (Natal)
4069
Rep. of South Africa

Dear Mr Fiddler,

We thank you for your interesting letter, dated April 18, 1989 and hereby gladly send you some material which we hope will be beneficial to your work. Enclosed is a set of brochures and technical guides that hopefully will answer you questions.

As you may notice the heat recovery from Wartsila Vasa engine is easy to accomplish, due to the high temperature level in the high temperature circuit of the engine. The temperature of the high temperature circuit will in all cases be above 90°C. By dividing the cooling system to low temperature and high temperature system heat recovery can be optimized.

To further increase the available waste heat an option with a two staged charge air cooler is available. In this case roughly 2/3 of the heat amount in the charge air cooler will be released to the high temperature circuit. In this way the available waste heat in the HT-circuit is nearly doubled on 100% load.

To describe these features we have enclosed the Wartsila Vasa 32 Project Guide for marine applications. The cooling system is described on page 47 and forwards. On page 55 you can see the available waste heat in the high temperature circuit for an engine with two staged charge air cooler. On pages 10-23 are technical datasheets for the different engine sizes.

./..

Postal address	Office address	Telephone	Telex	Telecopier
Oy Wärtsilä Ab Diesel Division P.O. Box 244 65101 VAASA, FINLAND	Pitkäkatu 2-12 65100 VAASA, FINLAND	+358-61-242 111	74250 wva sf	+358-61-111 906

Diesel Technology
K.Jofs/MMR 11 May 1989

We have also enclosed the standard engine brochures for the Vasa 22, Vasa 32 and Vasa 46 engines, as well as one brochure on fishing vessel and one brochure on cargo ship propulsion. Please note on page 11 on the cargo ship brochure is a chapter about waste heat recovery systems with some examples. Also please find enclosed an article about a typical freezer trawler of apparently similar size, equipped with a 6000 HP Wärtsilä Vasa 12V32 engine.

We hope that this information is sufficient for you. If you have any further questions please do not hesitate to contact us.

Kind regards

Oy Wärtsilä Ab
Diesel Technology

Kenneth Jofs
Technical Manager, Wärtsilä Vasa engines

ENCLOSURES:

- Wärtsilä Vasa 32 Project Guide
- Wärtsilä Vasa 22, Vasa 32 and Vasa 46 standard brochures
- Wärtsilä Vasa Diesel Engines for Propulsion and Power Production Onboard Fishing Vessels
- Cargo Ship Propulsion Viewpoints and Examples
- An article from Fishing News International - January 1988

P O Box 38027
Point (Natal)
4069
Rep. of South Africa.
18 April 1989

WARTSILA VASA FACTORY
TECHNICAL DEPARTMENT (DIESEL DIVISION)
P O BOX 244
SF - 65101 VAASA 10
FINLAND

Dear Sir,

I am a Marine Engineer presently employed as a Ship Surveyor (Engineer) and am in the process of preparing a thesis on energy saving in large freezer trawlers,for a Masters Diploma in Mechanical Engineering.

During the proposal preperation and literature review I have read with interest your article entitled "What makes a Fisherman's Engine". (World Fishing - February 1988 P22 - 23)

Although my study will be applicable to most large freezer trawlers, I am concentrating on a particular trawler fitted with a SULZER 6ZL 40/48 engine. I do feel however that your concept of HT and LT cooling water circuits as fitted to some of your WARTSILA main engines would also apply to the plant I intend to investigate.

I would be most grateful if you could furnish me with the details of your HT and LT cooling circuits and methods of heat recovery. Ialso intend using an exhaust gas boiler in the circuit.

Yours faithfully,

DM Fiddler.

D.M. FIDDLER

P.O. Box 38027
POINT (NATAL)
4069

7 April 1989

Mr. M. McWilliams
Superintendent Engineer
IRVIN and JOHNSON Ltd
(Trawling Division)
P.O. BOX 7444
ROGGEBAAI 8012

Dear Sir,

Further to our telephone conversation on the 4 April 1989 I would like to confirm that I will be in Cape Town from the 10 - 14 April 1989.

I would also like to take the opportunity of thanking you for all the information you have forwarded to me regarding the "ROXANA BANK" and allowing me access to the vessels logbooks and technical data.

My proposal states that I should gather data from the vessel while under actual operating conditions when the vessel is fishing. I would therefore be most grateful if you would allow me to sail in the vessel for part of the voyage (say approx. 10 days) and then transfer to one of your trawlers returning to Cape Town. As I should be at that part of my proposal by then, any time after the 15 June 1989 would suit me. My attendence of the vessel would be in a private capacity only.

From my written proposals syllabus, the project promises to be an interesting exercise in energy saving and your interest in my project is much appreciated.

Yours faithfully

DM Fiddler.

D.M. Fiddler.

P.O. Box 38027
POINT (NATAL)
4069

7 April 1989

SULZER BROS. (SA) Ltd
P.O. Box 2007
CAPE TOWN 8000

Your Ref: RFM/fa/4012

Dear Sir,

I have recieved some very informative data from your principals in Winterthur. The data is most relevant to the preparation of my dissertation on trawler energy saving.

I am most grateful for your assistance in obtaining this data for me and your interest is much appreciated.

Yours faithfully

D.M. Fiddler.

P O Box 38027
Point (Natal)
4060

Mr F. Müller
Head of Diesel Division
Sulzer Bros., (SA) Ltd.,
P O Box 2007
CAPE TOWN
8000

06 February 1989

Dear Sir

I am a Ship Surveyor (Engineer) employed by the Department of Transport (Marine Division) and am based at the Durban Office.

This year I will be completing a research project in energy saving in Fishing Trawlers and success in this project will give me a Master's Diploma in Mechanical Engineering and ultimately the 1st Class Marine Engineer Officer (Special Grade) Certificate.

I am using an exisiting stern trawler fitted with a Sulzer 6ZL40/48 main propulsion unit and a VTR401 Turbocharger.

It would assist me tremendously if you could let me have test bed results of the above mentioned engine and turbocharger and any other related information on this machinery.

Yours faithfully

D.M. FIDDLER

SULZER BROS. (SOUTH AFRICA) LTD.
Registration No. 010/7245/06

SULZER

Diesel Department

P.O. Box/Posbus 2007
Cape Town 8000

3rd Floor/3de Vloer
Fleetway House/Huis
Martin Hammerschlag Way
Foreshore
Cape Town 8001
Tel. (021) 21-5650
Telex 5 26463
Fax (021) 21-5656

Mr. D.M. Fiddler
P.O. Box 38027
POINT
4060 Natal

Quote our reference / Meld ons verwysing RFM/fa/4012

Your reference / U verwysing

Date / Datum 10 Feb. 19

Dear Mr. Fiddler,

We confirm the receipt of your letter dated 6.2.89.

We have sent your request to our principals in Winterthur who will prepare the required documentation.
In the meantime we trust that we have been of assistance to you and wish you the best success with your studies.

With kind regards,

yours faithfully
SULZER BROS. (SA) LTD.

R.F. MUELLER

P.O. Box 38027
POINT (NATAL) 4069
Rep. of SOUTH AFRICA

7 April 1989

SULZER BROTHERS Ltd
CH - 8401 WINTERTHUR
SWITZERLAND

Your Ref: DM 0721/Hu/Vo

Dear Sir,

I refer to your letter dated 22 February 1989 which accompanied the "Prototype Test" of your 16ZV40 propulsion engine, technical data on anciliary equipment and pamphlets on your new range of engines.

I am most grateful for the data recieved which will be of great assistance to me in preparing my dissertation on energy saving in fishing trawlers. Your interest is much appreciated.

The owners of the stern trawler have made available to me their shop trial records.

Yours faithfully

D.M. Fiddler.

GEBRÜDER SULZER AKTIENGESELLSCHAFT WINTERTHUR SCHWEIZ

SULZER

Mr D.M. Fiddler

P.O. Box 38027

ZA-4060 Point (Natal)
South Africa

Telegrams
Gebsulzer Winterthur

Please quote our reference in correspondence and telephone calls

Your reference	Your letter of	Our reference	8401 Winterthur
		DM 0721/Hu/Vo	22.02.89
	Direct dialing Phone	Direct dialing Telex	Direct dialing Telefax
	052/812368	89606070	052/224917

Fishing Trawler with a Sulzer 6ZL40 Main Engine

Dear Sir,

we refer to your letter dated 6 February, 1989, addressed to our office in Cape Town and are sending you attached the "Prototype Test" results of a 16ZV40 propulsion engine, taken at our test bed. For the shop trial records we recommend that you contact the owner of the stern trawler you are studying; they should be available on board of said vessel. We are also enclosing general technical data of ancillary equipment like pumps, coolers, etc.

Regarding information on the BBC turbocharger, please contact BBC/ABB directly (or their local representative), giving them the type of engine (with power and speed particulars) and type of turbocharger.

For your information we are also adding some pamphlets on our new range of engines.

Wishing you success with your studies we remain

yours faithfully,

SULZER BROTHERS
Limited

M. Huber

Gebrüder Sulzer Aktiengesellschaft CH-8401 Winterthur, Schweiz Telefon 052 - 81 11 22
Sulzer Frères Société Anonyme CH-8401 Winterthur, Suisse Telex 896 060 szch
Sulzer Brothers Limited CH-8401 Winterthur, Switzerland Telefax 052 - 23 42 92

P.O. Box 38027
Point
Natal
4069

Mr M. McWilliams
Superintendent Engineer
Irvin & Johnson Ltd
P O Box 7444
Roggebaai
8012

23 January 1989

Dear Sir

ROXANA BANK - ENERGY SAVING

1. I have almost completed my proposal which has to be forwarded to the Board of Examiners. I am also starting to make progress with the proposed thesis.

2. Unfortunately, due to Technikon committments I am unable to come to Cape Town in the immediate future to see the vessel and glean some more information.

3. Could you therefore please pass on the following information.

    3.1 Exhaust gas temperature (max. operating).

    3.2 Supply voltage.

    3.3 Does the fishmeal boiler gets its supply steam from the Aux. Boiler and if so, does the burner cut in and out during operation.

    3.4 The shaft rotational speed.

    3.5 The G.R.T.

    3.6 Place and year of build.

    3.7 Is there a F.W. generator on board.

...2/-

4. If it is available, I would appreciate a copy of the General Arrangement of the Engineroom.

5. How did you get your instruction in Plant Engineering for the G.C.C.? If there are no formal lectures is the only way through Veasey's Engineering College. Your comments would be appreciated.

Many thanks for all the information you have already passed on to me.

Yours faithfully

D.M. FIDDLER

Mr D.M.Fiddler
Ship Surveyor (E)
P.O. Box 38027
POINT, NATAL
4069

16 November 1988

Dear Sir

PROPOSED THESIS ON ENERGY SAVING

Further to your letter dated 10 November 1988, I enclose the following information :

3.1 Main Engine : Type Sulzer 6ZL 40/48, 4-stroke non-reversible, single acting.
T/C 1 x VTR 401

3.2 Fuel consumption : 156g/H.P./hr + 5% (diesel oil).

3.3 Auxiliary engine : Type Sulzer 5AL25, 2 off. Output 750 B.H.P. (Sulzer manual states 185 h.p./cylinder : 5 cyliner engine).
Each engine driving a 630kVA, 50Hz alternator.
Harbour set : Deutz F6L912, 45kW coupled to a 52kVA alternator.

3.3.1 Fuel consumption of Sulzer 5AL25 : 168g/B.H.P./hr.

3.4 Total Electrical Load :

(i) Total power supplied by 2 x 630 kVA alternators plus 2 x 700kVA shaft alternators.

(ii) Electrical load requirements as follows :

- Vessel steaming : require 1 x 630kVA alternator which would be running at 78% load.
- Vessel manouevring : 1 x 630kVA alternator at 84% load.

- Vessel fishing : 1 x 700kVA alternator supplying fishmeal plant and reefer plant at 90% load;
  1 x 700 kVA alternator supplying trawlwinch converters from zero to 90% load;
  1 x 630kVA alternator supplying ships load at 66%.
- Vessel discharging : require 1 x 630kVA alternator at 82% load.

3.5 <u>Reefer plant capacity</u> : 3 x screw compressors at 155000kcal/hr
1 x screw compressor at 590000kcal/hr.

<u>Freezing Hold No. 1</u>	+- 922 cu.m
<u>Freezing Hold No. 2</u>	+- 638 cu.m
<u>Freezing Hold No. 3</u>	+- 450 cu.m (fishmeal only). Independent cooling source to maintain temperature at +12° C.

With regard to holds 1 & 2 : we use a factor of 0,7 for stowage, therefore working on 700kg/m^3. Hold temperatures maximum -30° C (no's 1&2).

3.6.1 Flexible coupling between engine and gearbox.

3.6.2 2 x 700kVA alternators, one port and one starboard, with a "Vulcan" flexible coupling between gearbox and alternator. Method of engagement : oil operated clutch for each shaft.

3.7 <u>Auxiliary Boiler Type VL</u> Oil fired side : output 2500kg/hr. Fired by a burner unit of +-200kg/hr.
Main engine exhaust side : output 800kg/hr.

3.8 <u>Fishmeal Plant Details :</u>

Type FM50B.

Raw fish processing capacity : 50 - 60 tonnes / 24 hrs.

Fish meal production capacity : 8 - 12 tonnes / 24 hrs.

Fish oil production capacity : 0,8 - 3 tonnes / 24 hrs (not in use).

Power of electric motors installed : 113,2 kW.

Heating steam consumption : 1200 - 1300 kg/hr.

Heating steam pressure : 4 - 6 kg/cm^3.

3.9 1 x 700kVA alternator to supply power as follows : 2 winches aft, one port, one starboard. Each drive as follows : A.C. motor driving D.C. generator driving D.C. motor on winch. Varying volts on D.C. generator varies speed of motor (Ward-Leonard).

1 x net winch driven by an A.C. motor.

3.10 <u>Main Engine :</u> Fuel is blended, 457 secs Redwood No.1 at 100° C (38° C).

<u>Auxiliaries:</u> Gas oil

With regard to 3.2 above, the following figures were recorded whilst on blended fuel :

July = 298,44 l/hr = 7162,59 l/day.

Sept = 347,79 l/hr = 8346,99 l/day.

I hope that the above information is adequate for your requirements, but should you require any further details, please avail yourself of the opportunity to contact me as required.

I would also like to wish you every success on your thesis.

Good luck!

Yours faithfully

M. McWill

M. McWILLIAMS

P O Box 38027
Point, Natal
4069

10 November 1988

Mr M. McWilliams
Engineer Superindent
Irvin and Johnson Limited
P O Box 7444
ROGGEBAAI
8012

Dear Sir

Re : <u>PROPOSED THESIS ON ENERGY SAVING ON MFV "ROXANA BANK"</u>

1. Further to the letter dated 28 September 1988 from Mr G. Ralston, Engineering Manager, I require information in order to submit my proposal to the Board of Examiners at the Natal Technikon.

2. The proposal will consist of a problem statement, what I propose to do and a breakdown of the vessels machinery and auxiliary equipment.

3. Please forward to me at your convenience the following information :-

   3.1 Main Engine type, number installed and power rating;

   3.2 Main Engine fuel consumption;

   3.3 Auxiliary Engines type, number installed and power rating;

      3.3.1 Auxiliary engine fuel consumption;

   3.4 Total electrical load;

   3.5 Reefer plant Capacity;

   3.6 Gearbox particulars :-

      3.6.1 Main engine connections (method);

      3.6.2 Gear driven generator/alternator if fitted;

      3.6.3 Any other power take offs;

   3.7 Auxiliary boiler type (if fitted);

   3.8 Fish meal plant details (basic);

   3.9 Power supply for trawl winch;

   3.10 Grade of fuel used - Main Engines, Auxiliary Engines.

4. I will be in Cape Town sometime in January 1989 and would like to visit the ship when she is in port. I will plan my itinery to coincide with the vessels stay at your berths. Please furnish the programme for the vessel in January/February.

5. Should you wish to communicate the above information to me telephonically, I can be reached at (031) 3071501/2/3/4 or through Mr Butcher of the Cape Town Department of Transport. I would like to submit my proposal at the end of December 1988.

6. I would like to thank you for the assistance given to me in arranging the above information.

Yours faithfully

D M Fiddler

....

D.M. FIDDLER
Ship Surveyor (E)

P O Box 38027
Point, Natal
4069

10 November 1988

Mr G. Ralston
Engineering Manager
Irvin and Johnson Limited
P O Box 7444
ROGGEBAAI
8012

Dear Sir

Your letter dated 28 September 1988 refers.

1. I undertake to keep all information on the "Roxana Bank" confidential.

2. The only other people who will see the data will be my project leader - a Senior Lecturer at the Natal Technikon and the examiner - as yet unnamed.

3. The terms of my study contract with the Department of Transport (Marine Division) all research is to remain confidential.

Yours faithfully

D M Fiddler

....

D.M. FIDDLER
Ship Surveyor (E)

EAST QUAY/OOSKAAI
DOCKS/DOKKE
CAPE TOWN/KAAPSTAD
TEL. AD. "TRAWLER"
TELEX 5-27622
TELEFAX (021) 419-3738
TEL. 25-1300
P.O. BOX/POSBUS 7444
ROGGEBAAI 8012

A MEMBER OF THE ANGLOVAAL GROUP
LID VAN DIE ANGLOVAAL GROEP

REG. NO. 52/01693/06

TRAWLING DIVISION
VISTREILAFDELING

IRVIN & JOHNSON LIMITED

GR/BH
28.9.88

Mr D M Fiddler
P O Box 38027
POINT
4069

Dear Sir

Further to your letter of 21.08.88 concerning your thesis for your Masters Diploma, we would be happy to supply you with information on the vessel, Roxana Bank.

We must however have an undertaking from you that vessel's name and information supplied be treated in strictest confidence.

We suggest you contact Mr M McWilliams, the vessel's superintendent, for any information you may require.

We wish you success in your studies and look forward with interest to receiving a copy of your thesis.

Yours faithfully
IRVIN & JOHNSON LIMITED
TRAWLING DIVISION

G RALSTON
ENGINEERING MANAGER

cc: J Steward
General Manager

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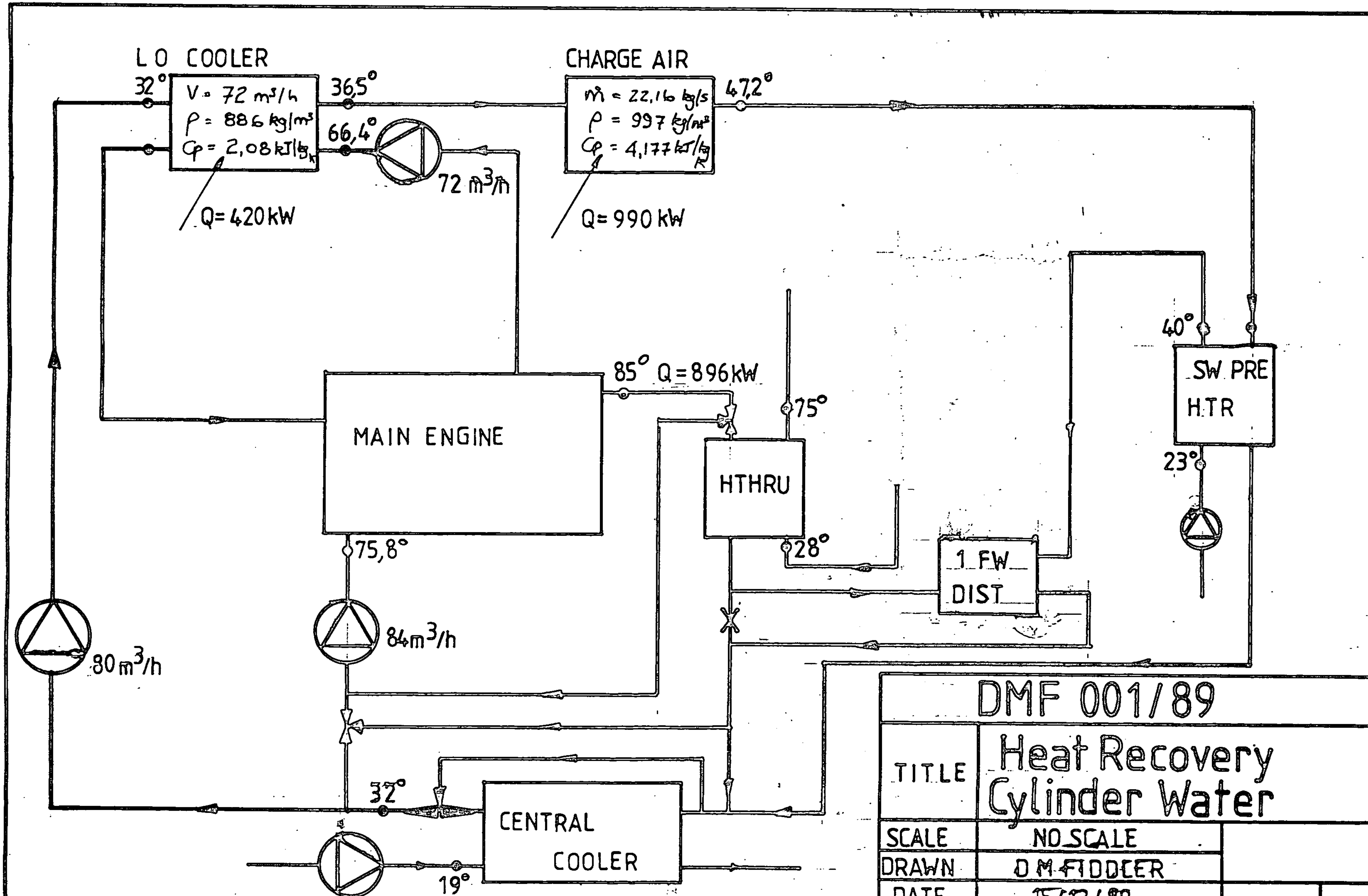
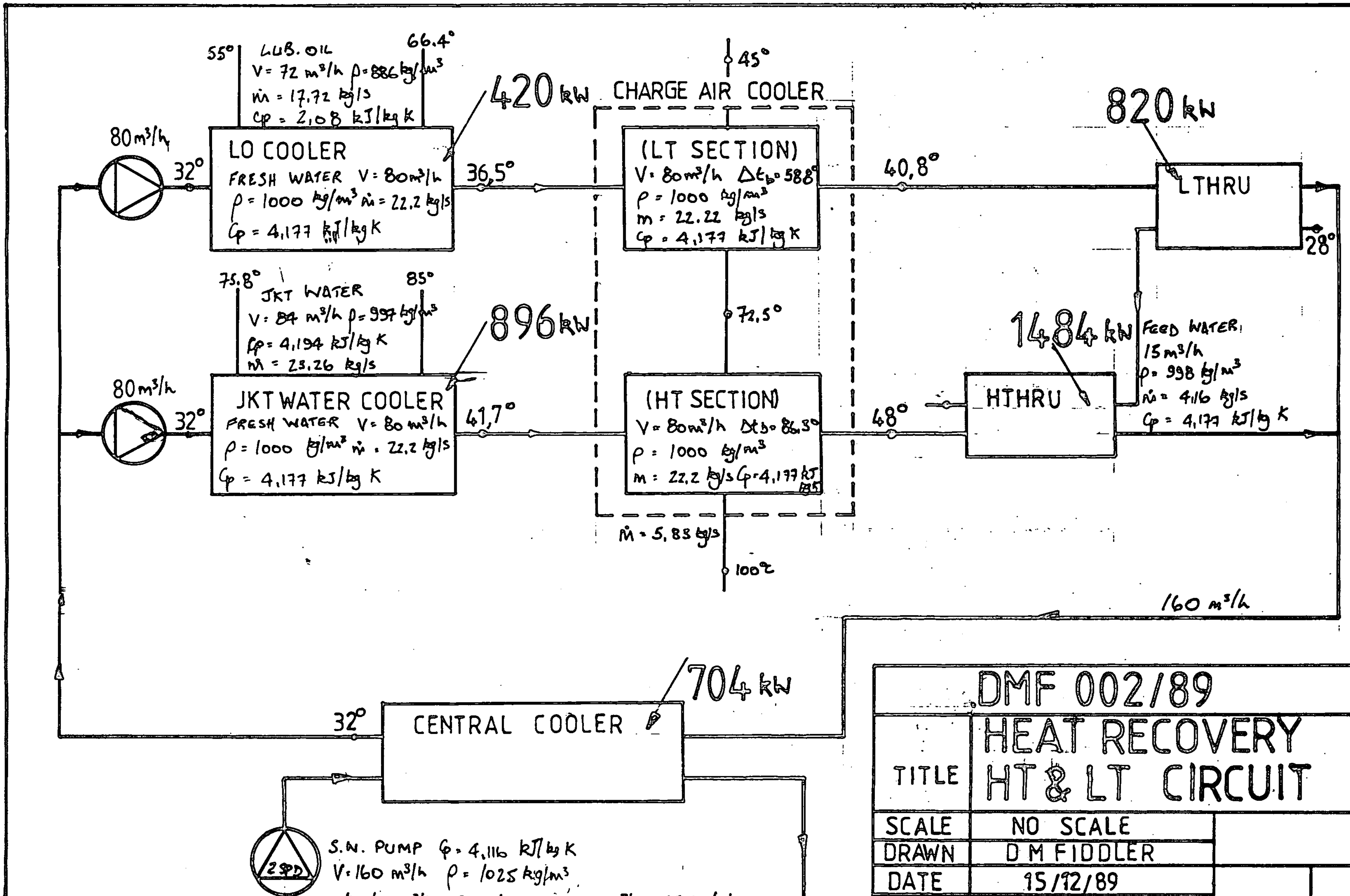
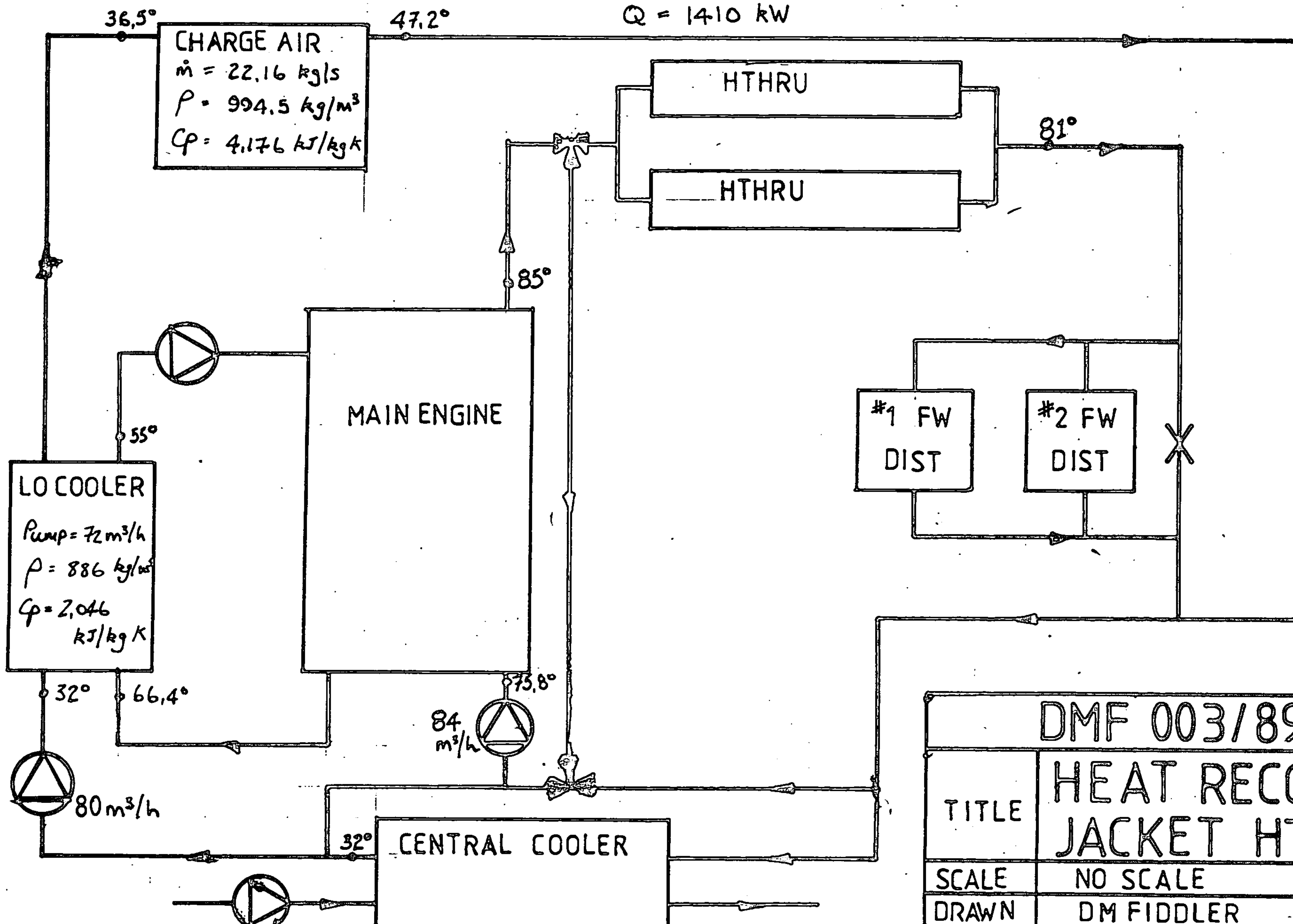


L O COOLER
32°
V = 72 m³/h
ρ = 886 kg/m³
Cp = 2,08 kJ/kg K
36,5°
66,4°
72 m³/h
Q= 420 kW
CHARGE AIR
ṁ = 22,16 kg/s
ρ = 997 kg/m³
Cp = 4,177 kJ/kg K
47,2°
Q= 990 kW
40°
SW PRE
H.TR
23°
MAIN ENGINE
85° Q=896 kW
75°
HTHRU
28°
75,8°
1 FW
DIST
84 m³/h
80 m³/h
32°
CENTRAL
COOLER
19°
DMF 001/89
TITLE
Heat Recovery
Cylinder Water
SCALE
NO SCALE
DRAWN
D M FIDDLER
DATE
15/12/89



LUB. OIL
V = 72 m³/h ρ = 886 kg/m³
ṁ = 17.72 kg/s
Cp = 2,08 kJ/kg K
55°
66.4°
80 m³/h
32°
LO COOLER
FRESH WATER V: 80 m³/h
ρ = 1000 kg/m³ ṁ = 22.2 kg/s
Cp = 4,177 kJ/kg K
420 kW
36,5°
CHARGE AIR COOLER
45°
(LT SECTION)
V: 80 m³/h Δt = 5.88°
ρ = 1000 kg/m³
m = 22.22 kg/s
Cp = 4,177 kJ/kg K
72,5°
40,8°
820 kW
LTHRU
28°
75.8°
JKT WATER
85°
V: 84 m³/h ρ = 997 kg/m³
Cp = 4,194 kJ/kg K
ṁ = 23.26 kg/s
896 kW
80 m³/h
32°
JKT WATER COOLER
FRESH WATER V: 80 m³/h
ρ = 1000 kg/m³ ṁ = 22.2 kg/s
Cp = 4,177 kJ/kg K
41,7°
(HT SECTION)
V = 80 m³/h Δt = 86.3°
ρ = 1000 kg/m³
m = 22.2 kg/s Cp = 4,177 kJ/kg K
ṁ = 5,83 kg/s
100°
48°
HTHRU
1484 kW
FEED WATER
15 m³/h
ρ = 998 kg/m³
ṁ = 4.16 kg/s
Cp = 4,177 kJ/kg K
160 m³/h
704 kW
32°
CENTRAL COOLER
2 SPD
S.W. PUMP Cp = 4,116 kJ/kg K
V: 160 m³/h ρ = 1025 kg/m³
DMF 002/89
TITLE HEAT RECOVERY HT & LT CIRCUIT
SCALE NO SCALE
DRAWN D M FIDDLER
DATE 15/12/89



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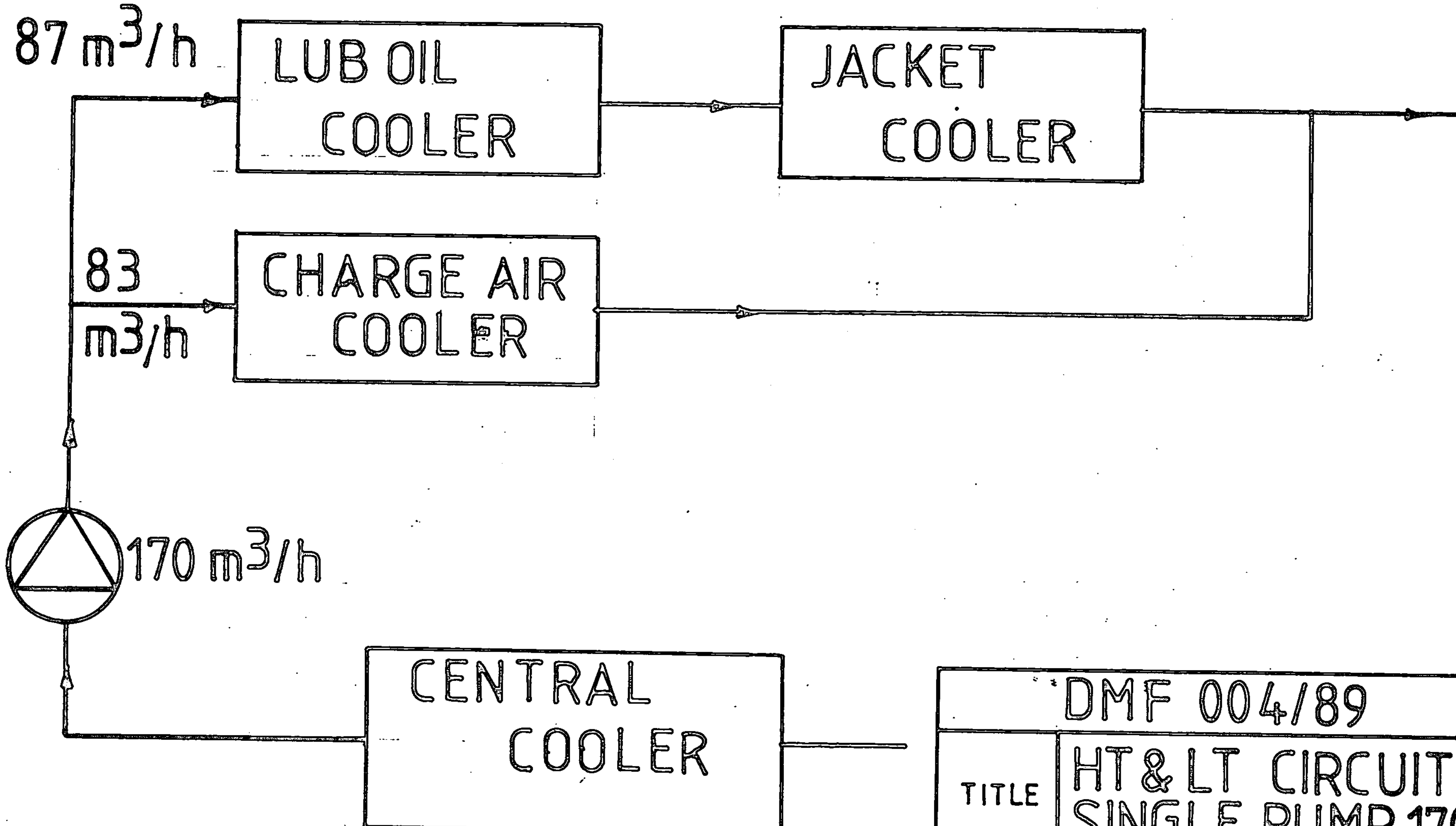
DATE	15/12/89



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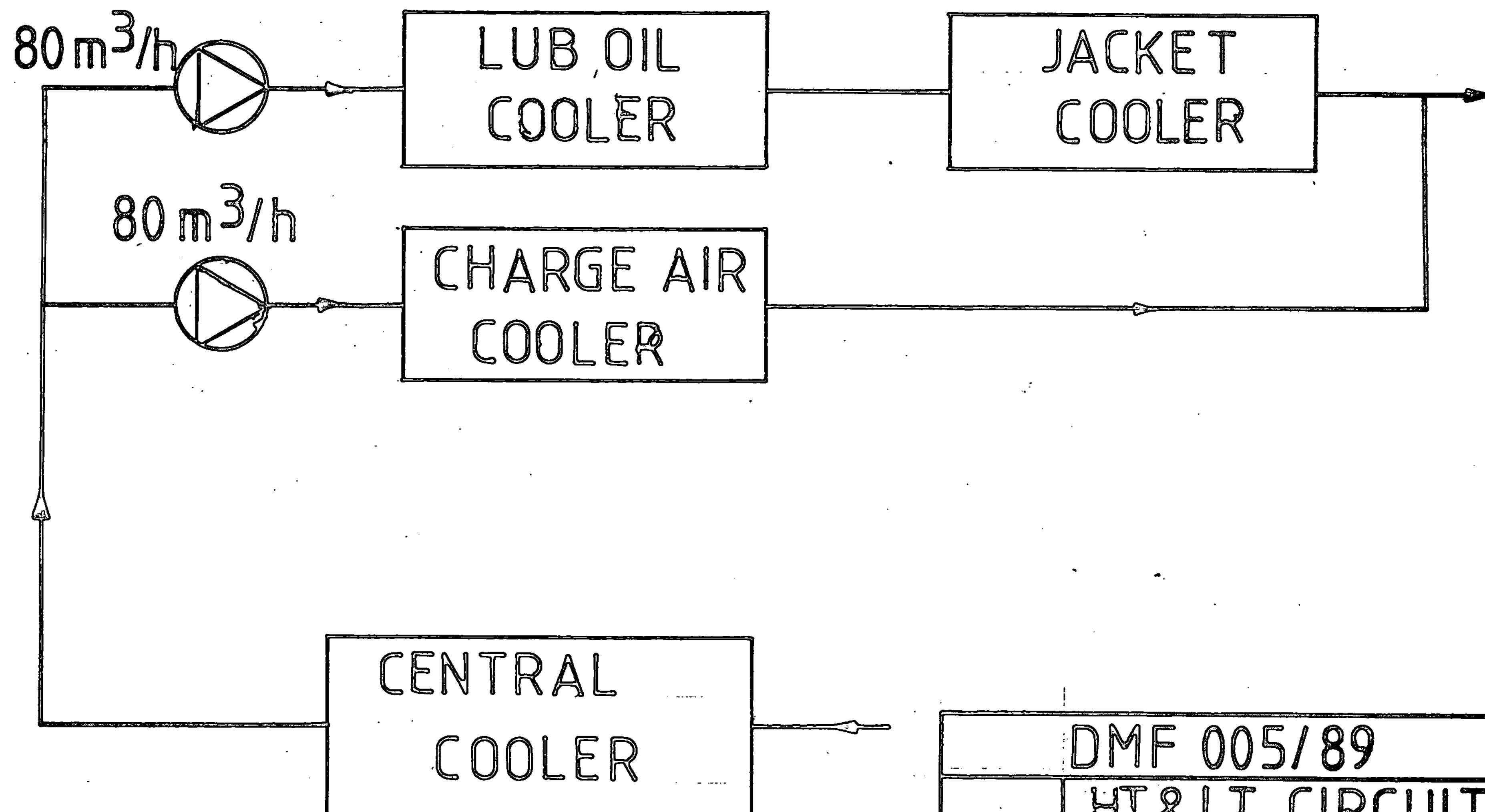

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SCALE	NO SCALE
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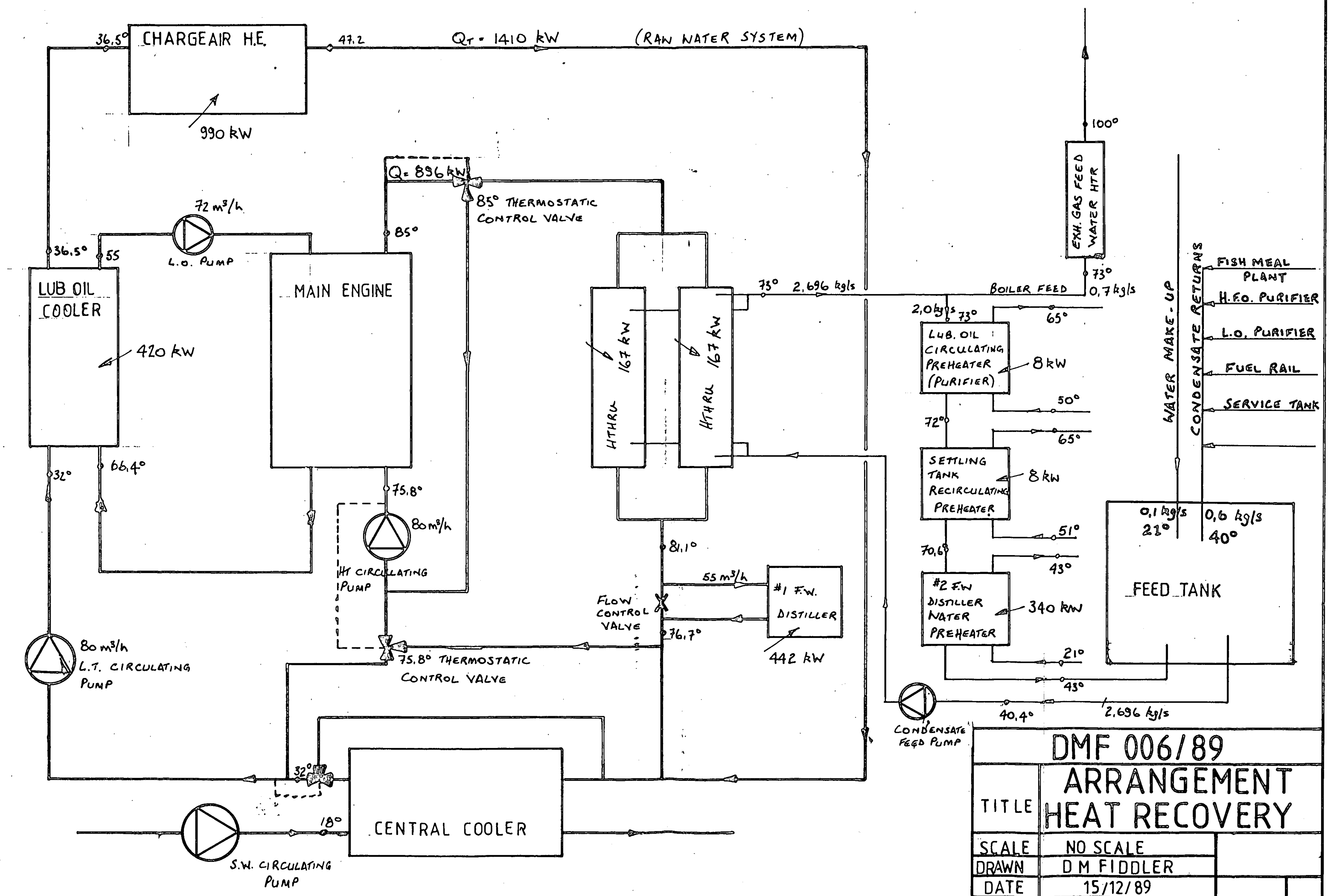
DATE	15/12/89



DMF 006/89

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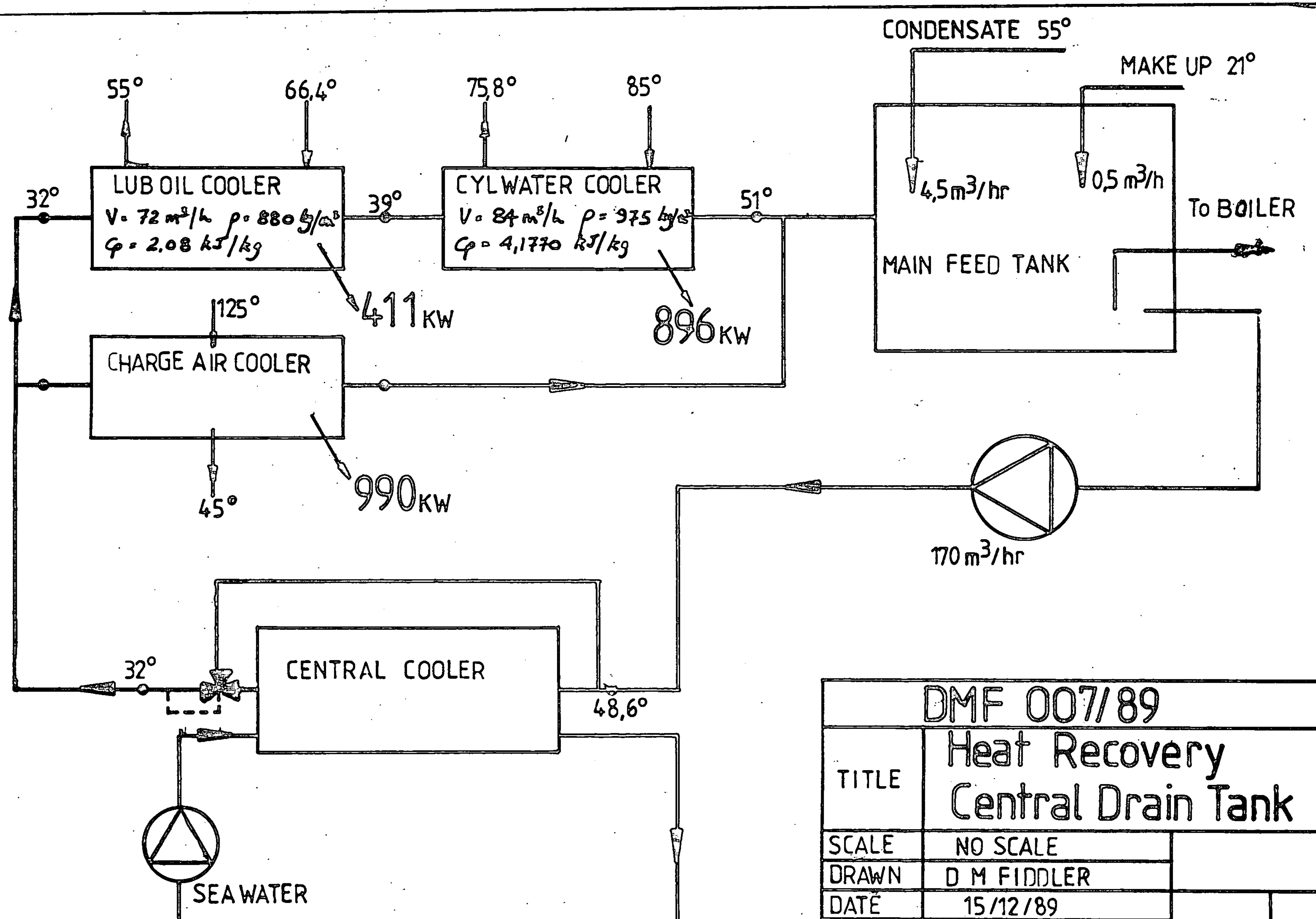
DATE	15/12/89



DMF 007/89		
TITLE	Heat Recovery Central Drain Tank	
SCALE	NO SCALE	
DRAWN	D M FIDDLER	
DATE	15/12/89	