Power Condenser Heat Transfer Technology

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POWER CONDENSER HEAT TRANSFER TECHNOLOGY

Computer Modeling / Design / Fouling

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HEMISPHERE PUBLISHING CORPORATION
Washington New York London

McGRAW-HILL BOOK COMPANY
New York St. Louis San Francisco Auckland Bogotá
Hamburg Johannesburg London Madrid Mexico
Montreal New Delhi Panama Paris São Paulo
Singapore Sydney Tokyo Toronto

Library of Congress Cataloging in Publication Data
Main entry under title:

Power condenser heat transfer technology.

"Proceedings of the workshop titled Modern developments in marine condensers, held at the Naval Postgraduate School, Monterey, California, March 26-28, 1980, and sponsored by the Naval Sea Systems Command and the Office of Naval Research"—Verso t.p.

Includes bibliographical references and index.

1. Condensers (Steam)—Congresses. 2. Heat—Transmission—Congresses. 3. Marine engines—Congresses. I. Marto, P. J. II. Nunn, R. H. III. United States. Naval Sea Systems Command. IV. United States. Office of Naval Research.

TJ557.P68 621.1'97 81-4540
ISBN 0-07-040662-6 AACR2

Proceedings of the workshop titled "Modern Developments in Marine Condensers" held at the Naval Postgraduate School, Monterey, California, March 26-28, 1980, and sponsored by the Naval Sea Systems Command and the Office of Naval Research.

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POWER CONDENSER HEAT TRANSFER TECHNOLOGY

ᴾreface

The surface condenser, since the first experiments by James Watt in 765, has undergone a considerable evolution in efforts to improve upon ower plant performance. This development has paralleled the growth of the se of steam for electric power and marine propulsion, and has required the fforts of many outstanding scientists and engineers, including James rescott Joule, Osborne Reynolds, and Wilhelm Nusselt. Today, surface conensers exist in all shapes and sizes, but unfortunately their designs have een based largely upon historical and empirical information.

In recent years, technical advances have occurred in two important areas at could and should impact significantly upon surface condenser designs of e future. In the first instance, it is well known that the advent of largeorage-capacity digital computers has made it possible to solve a wide riety of complex engineering problems in a reasonable time period. Such mputer solutions, when coupled with the latest numerical optimization hemes, can provide the engineer with a most sophisticated design tool. In dition, the heat transfer community is well aware of the numerous publitions in recent years that have treated the subject of heat transfer enhanceent. Many appealing paths consequently exist that can be followed to hieve improved heat transfer during condensation, or single-phase forced nvection. What remains to be done now is to utilize these two advances gether to provide for a compact, yet efficient, surface condenser that will m part of tomorrow's advanced steam power systems. This objective is ticularly important for naval vessels where, in addition to cost, machinery e and weight are critical.

This volume attempts to bring these vital areas together, along with other damental developments pertaining to condenser-related phenomena. It resents the proceedings of a workshop titled "Modern Developments in rine Condensers," held at the Naval Postgraduate School, Monterey, Calinia, March 26-28, 1980, under the sponsorship of the Naval Sea Systems nmand and the Office of Naval Research. The workshop was attended by ividuals from government, universities, and industry, from the United tes and abroad, who have had an active involvement in the advancement the technologies related to marine surface condensers. Topical areas inle: computer modeling, the effects of noncondensable gases, vapor shear condensate inundation, enhancement, and fouling. Each of these topics eated by a keynote paper followed by prepared discussions and an open ussion. A final panel discussion is provided to summarize important ings.

This volume provides an overview of current condenser technology, tifies problem areas for further research, and establishes improved design niques. It should be useful to design engineers interested in improving er system performance and to research scientists interested in studying amental phenomena that occur within a surface condenser.

We are deeply indebted to many people for their outstanding efforts in assisting us with this endeavor. In particular, we would like to thank the authors of the keynote papers and the invited discussers for their diligence in responding to our many requests. We also want to thank each of the participants for the lively discussion that developed during the workshop, much of which has been reproduced in this volume. Finally, our sincere thanks go to Mrs. Vicki Culley for her invaluable assistance in organizing all the details of the workshop, and to Mrs. Jill Hutcheson for painstakingly typing much of the manuscript and transcribing the discussion tapes with both professional aplomb and a cheerful spirit.

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Introduction

Introduction

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Steam propulsion has been an unfashionable subject for the last few decades. The trend for merchant ships towards diesel engines, and for warships (with the notable exception of submarines and other nuclear vessels) towards gas turbines, made steam machinery appear rather Victorian. However in recent years the trend has begun to reverse, thanks to the 'Energy Crisis'. On the one hand, a steam boiler is one of the easiest ways of extracting energy from difficult fuels such as coal or heavy residual oil, and on the other hand an unfired steam generator is an attractive means for recovering waste heat from the exhaust of gas turbines or diesels in COGAS or CODAS systems. There is therefore renewed interest in reducing the bulk and weight of steam machinery, including the condenser.

Meanwhile the design experience embodied in design codes such as that of the Heat Exchange Institute (HEI), and later more sophisticated computer models, relates mainly to condensers for land-based power stations where the object is to extract the maximum energy from the steam (by providing the lowest possible sink temperature) consistent with minimum capital and running costs. This is achieved by designing for a high vacuum and a small steam pressure drop, minimizing the water temperature rise and pumping power, and using inexpensive materials. Generous margins are provided to ensure the achievement of the thermal performance and the designer is able to assume nearly constant steam and water conditions, with the condensers normally operating near full power. Bulk and weight are not of great importance. These criteria are in contrast to many of the conditions for which condensers are designed for marine and naval use, and particularly for submarine application.

PHYSICAL DIMENSIONS

Naval condensers are an order of magnitude smaller than land-based power condensers: their duty is measured in tens of megawatts, rather than hundreds. Their actual dimensions have a critical bearing on their effectiveness in relation to the ship's cost and performance. Compactness is more important than thermal effectiveness when the overall performance of the ship is considered and in a submarine the diameter of the pressure hull can depend on the depth of the condensers below the turbine shaft line.

An extra half metre on the diameter or metre on length can have a far larger adverse effect on the cost and performance of the whole ship than the loss of a few tens of millibars and this applies also, though to a more limited extent, to surface ships. These considerations lead to the acceptance of modest vacua, high surface loading (typically 0.055 kg/m^2s at 0.2 bar) and moderate pressure drops. They also permit the use of relatively costly tubes of exotic geometry, designed to operate at high heat fluxes, though at a penalty in cooling-water pumping power. Narrow margins on design are permissible. The penalty for under-design is a small reduction in maximum power; but, because of the cubic relationship between power and speed, a deficiency of a few tens of mb of vacuum at full power may mean a loss of only a fraction of a knot on full speed, which the vessel may only need for a small fraction of its total life. The penalties for over-design, however, are present all the time in the form of excessive bulk and weight, leading at best to a crowded machinery space with poor accessibility, and at worst to a boat of greater displacement and wetted area, and hence to a demand for more power, a still larger condenser, and so on in a vicious circle.

Naval condensers have a wide turndown, with only a small proportion of time spent at full power, though it is these full power conditions which govern design. If the condensers are over-designed for full power, there are problems in vacuum control at low power to avoid undercooling, oxygenation of condensate, and erosion from excessive wet steam velocities.

VARYING CONDITIONS OF SERVICE

Apart from this wide turndown, submarine condensers are subjected to high water pressure (though not continuously), to the changes in attitude of the ship, and to large variations in water temperature. The overriding necessity to preserve the integrity of the water-side as a pressure vessel affects the mechanical design of tubes, water boxes, and ducting and has consequent effects on the thermal design. Circular section water boxes to accommodate the pressure determine the geometry of the tube bundle; high-strength, corrosion-resistant materials such as 70:30 cupro-nickel or titanium are chosen rather than cheaper materials of higher thermal conductivity. To minimize the size of the hull penetration, the design water throughput is kept low, which gives rise to high temperature increments (as high as 40K in some circumstances).

The movement of the ship allows a reduction in pump size and power: the circulating water can largely be provided by scoops. But because of the non-linearity between turbine power and water flowrate from the scoops (one has a cubic relationship with speed, and the other a quadratic relationship), there is a problem of excessive circulating water at low speeds. The pitching and rolling of the ship and the dive angle of a submarine must be taken into account in the design of the hot well to ensure adequate water head over the suction pump at all times, but without flooding the lower tubes. These considerations increase the height required.

Circulating water temperatures can vary from an arctic $0^{o}C$ to a tropical $30^{o}C$, which could give a two-to-one variation in l.m.t.d. If the condenser were designed for either extreme, there would be severe penalties at the other end of the range. It is normal to design for the temperate case, but it would be very desirable to be able to employ special techniques to boost the performance in tropical waters for short periods, for example by the injection of promoters of dropwise condensation.

Naval condensers form part of a simple steam cycle which is often without de-aerators or feed heaters. The condenser itself must also act as a de-aerator and measures must be taken to minimize undercooling. However, if condensate at minimum temperature is required for auxiliary cooling (for the air ejector condenser, for example), it may be desirable to segregate condensate from the cold (water inlet) end.

Fouling is less of a problem in mid-ocean than for land-based coastal sites and conventional 'fouling factors' are over-generous. But biological fouling of water boxes at low water throughput and the effects of polluted harbour waters must be borne in mine, and also the possibility of scaling by reverse-solubility salts under conditions of high water outlet temperature.

IMPLICATIONS FOR THERMAL DESIGN AND THE NECESSARY RESEARCH AND DEVELOPMENT

For the reasons mentioned above it is necessary to design naval condensers as near the bone as possible. It is not adequate to assume uniform conditions throughout the condenser with a l.m.t.d. and an overall coefficient which are assumed to be representative and with a generous 'fouling margin' or factor of ignorance. The penalties for over-design are too onerous: a more accurate knowledge of heat-transfer behaviour, particularly on the highly non-uniform steam side, is needed.

The HEI procedure, on which the USN's Design Data Sheet for condensers DDS4601-1 is based, calculates an overall heat transfer coefficient based on the square root of water velocity with constant multiplying factors to allow for changes in water inlet temperature and tube dimensions and materials from the assumed standard conditions, and for fouling. No account is taken of varying conditions on the steam-side, such as changes in steam density and velocity, condensation rate, air content and inundation. In a power station condenser this approach is justified because at water velocities below 2 m/s the water side resistance is controlling, and a small temperature rise and low surface loading gives approximately uniform steam-side conditions. However, the use of constant multiplying factors to calculate the effect of changing one resistance when the other may vary widely is basically unsound. For example, if the clean overall coefficient has been increased from 4 to 6 kw/m^2K by the use of thinner tubes and dropwise condensation, with no change to water side conditions, the water-side fouling is also presumably unchanged, but the implied fouling resistance corresponding to an 0.9 'fouling factor' has apparently decreased from 28 to 19 x $10^{-6}m^2K/W$. In this case the use of an added fouling resistance would be more appropriate than a multiplying factor: on the other hand if the change in clean overall coefficient has been due to changes on the water-side (increased velocity or turbulence), it has probably been accompanied by a reduced fouling tendency, and the use of a multiplying factor may be realistic.

Attempts to model the steam-side heat transfer normally start with Nusselt's theory of laminar condensate film drainage under gravity. This however, gives extremely over-conservative results for a highly rated condenser. High velocities in relatively high density steam give rise to large shear forces on the condensate film: high condensation rates per unit surface area give rise to high temperature gradients through the film, which may be highly turbulent and far removed from Nusselt's assumptions. Thickened films break up into sub-cooled droplets on which condensation can also take place. Because of this splashing, turbulence and direct-contact condensation on falling drops, the effect of inundation on lower tubes is less detrimental than the Nusselt model would imply.

Thus in the absence of non-condensible gas, the steam-side coefficient is high, but air blanketing at high condensation rate seriously degrades performance, and the correct location of air vents is of great importance. In general, air will tend to accumulate wherever it is not either 'pushed' by high velocity through flow of steam, or 'pulled' by a venting channel connected to the air ejector. In order to avoid overloading the air ejector with uncondensed steam, it is necessary to provide a section of air cooling tubes between the air accumulation zone and the point where the air-vapour mixture leaves the condenser, shielded from condensate carry-over and baffled to ensure that velocity over the tubes is reasonably high and that live steam cannot bypass into the exit duct or reheat the extracted air. The proportion of condenser area to be allocated to air cooling still needs optimizing: traditionally it has been of the order of 8-10%, but there is evidence that a smaller area can be effective.

A high condensing rate can theoretically permit some recovery of dynamic pressure because of the 'diffuser' effect of the decelerating steam and thus preferential flow through the cooled tube bundle rather than the empty lane. Moreover, high momentum in the steam can also cause the steam to flow preferentially through the bundle rather than follow a steam lane if the latter requires a change in direction. It seems possible that the optimum heat-transfer and pressure drop for a given tube-plate area might be obtained by varying the pitch of the tubes continuously rather than by adopting blocks of tubes of uniform pitch separated by steam lanes. This might allow the maintenance of high steam velocities (with beneficially high rates of shear and of mass transfer) as condensation proceeds, by providing a diminishing cross-section for flow. It suggests a radially inward flow with the air off-take at the center. However, if condenser width is critical for machinery space layout, there is little advantage in achieving say 5% improvement in heat transfer with wide side lanes permitting radial flow, at the cost of 10% increase in width to accommodate the lanes.

A high temperature rise in the cooling water implies that the steam flows three-dimensionally, with the mass flux at the cold (water inlet) end some three times greater than that at the hot end. A low water flowrate with a high velocity favours a two-pass arrangement, which happens to be more convenient with respect to the sea-water pipe system. If it is necessary to stack passes one above the other, a decision must be made as to whether the colder pass should be on top or below. Placing it on top permits rapid condensation of the bulk of the steam and minimizes pressure drop, but if there is a high rise in water temperature, it is also desirable to make use of the advantages of counterflow to maximize the effective mean temperature difference. This is particularly so where the steam side conditions are noticeably worse in the lower part of the condenser because of low velocity, high air content, and heavy inundation and where it is desirable to have the highest temperature difference.

Elimination of the heavy return-end header in a 2-pass condenser is attractive and can be achieved by the use of U-tubes. These are used in conventional high pressure condensing feed heaters, but here the bend constitutes only a small proportion of the total surface. Little is known about steam-side heat transfer and condensate film behaviour on the outside of a U-bend (probably in the vertical plane), and the water-side heat transfer pattern is complicated by the absence of half-way mixing and the varying lengths of tube.

ENHANCEMENT OF HEAT-TRANSFER

The importance of compactness justifies measures to raise the overall heat transfer coefficients of condenser tubes despite the increased pumping power and tube cost which may then arise. These measures can include the use of turbulence promoters on the water-side and improved condensate removal or extended surfaces on the steam-side. Whatever method is used there must be no risk to mechanical integrity by, for example, erosion damage, stress corrosion, or fretting, nor risk of tube blockage by fouling; and there can be no interference with normal maintenance. The last provision apparently rules out spiral inserts which might prevent cleaning or inspection of the tubes.

The use of titanium, because of its very high erosion resistance, permits the use of turbulence promoters and tight radius bends (where partial blockage by debris might occur), which could not be contemplated for more conventional materials.

Pressure drop per unit length of tube at a given velocity will inevitably be greater for advanced geometry tubing in which turbulence is promoted on the water-side, but if the design is optimized for the use of such tubes, with reduced condenser length, reduced velocity through the tubes and perhaps reduced mass flow through the whole system, the overall system pressure drop need not be increased. Possible problems to be guarded against if enhanced tubes are adopted include increased fouling deposition in the lee of protrusions, and generation of unacceptable noise and vibration.

Because the steam-side and water-side resistance are of the same order, an enhancement in either is beneficial, though ideally both should be improved simultaneously. The optimum geometry for the air-cooling region is different from that for regions of heavy condensation where, for example, fins would be of little use because of flooding between them caused by the high surface tension of the condensate.

Dropwise condensation is attractive whether it is achieved by a permanent, thermally-thin coating or by injection of promoter. With a condenser designed say for 80% of full power in temperate waters, injection could be used to provide boosting to full power and in tropical waters. Boosting for full power would need to be almost instantaneous, but boosting for the tropics could be more leisurely. Unfortunately, however, promoters which have appeared satisfactory in laboratory tests on isolated tubes are nearly always disappointing in practical trials. Such failures are commonly attributed to inadequate dispersion or adherence of the promoter, oxidation, fouling etc., but could be due to the relative unimportance of the resistance of the condensate film to heat-transfer when falling drops of subcooled condensate present a large surface for direct-contact condensation, particularly in regions of high air concentration.

Another technique for boosting performance is the partial jet condenser. The main condenser and its circulating water system is under-designed for say 90% duty, but with an oversized extraction pump: at 100% load the excess steam is condensed on a spray of cooled condensate recirculated to the top of the shell. The additional heat exchanger for rejecting heat from this spray water to the sea can be located anywhere convenient: possibly outside the pressure hull with scoop circulation of sea water.

CONCLUSIONS

There is an evident need for more information about steam-side phenomena in highly rated condensers, and no lack of problems for this Workshop to discuss. Experimental validation of computer simulation models is still lacking, and because of the complex 3-dimensional flow and heat and mass transfer patterns in the 2-phase steam-air-water mixture, experiments in large scale rigs are required as well as in the small scale rigs such as those in use here at NPS. At the Admiralty Marine Technology Establishemnt at Portsmouth (England) we have a 2-pass experimental condenser of over $100m^2$ surface area, with instrumentation on individual tubes to measure water temperature rise and local vapour temperature, so that patterns of heat transfer and air accumulation can be mapped, and with its help we are investigating different arrangements for air extraction, varying width of side lanes, advanced geometry tubes, de-aerating sprays etc.

Some of the questions which the Workshop might consider:

Should radial flow be encouraged with generous side lanes, or should tube bundles be close-packed with high velocity down-flow? Can pressure recovery be achieved at high condensation rates?

Should condensate be removed as soon as possible after it is formed, or is its descent through the bundle as a heavy rainstorm beneficial?

Where is the best location for air vents, how can air bubble accumulation be avoided, and what proportion of condenser surface should be devoted to air cooling?

Is steam distribution at inlet to the condenser important, and how can it best be measured? (Laser anemometry?)

Is there any advantage in departing from standard tube diameters and P/D ratio?