Heat Losses from Oil-tanker Cargoes

R. J. SAUNDERS, C.Eng., A.M.I.Mech.E. (Member)*

Tests were carried out on two tankers carrying heavy crudes to determine heat losses from the cargo tanks under typical service conditions. Instrumentation was fitted in a wing tank of each vessel to measure the heat losses, shell-plate and bulk-oil temperatures and temperature gradients at the tank bottom and sides. Measurements were made while the oil was allowed to cool and during reheating to the arrival temperature.

It was found that for the immersed areas of the tank the shell-plate temperatures exceed the sea temperature by only a few degrees and that steep temperature gradients exist on the oil side of the plate. The stagnant films forming under these conditions account for the relatively small losses occurring from cargoes of hot viscous oil. With no steam on the heating coils the heat lost from the tank bottom was almost negligible. Heat losses from the deck were subject to wide fluctuations, including reversal of direction, according to ambient conditions. Losses from the topsides also varied considerably, but in general could be equated with those for the wetted sides. Heat lost per ft^2 from the wetted sides was on average more than double that from the deck or bottom of the tank, heat-transfer coefficients covering the worst conditions experienced being as follows:

	Btu/ft^2 h °F
Deck	1·5
Sides	3·3
Bottom	1·5
Internal bulkheads to empty wing tanks	0·8

Average values can be used for design purposes instead of these maxima, if desired.

The significance of the high coefficient for the ship sides is that the heat lost from a wing tank so far exceeds that from a centre tank, volume for volume, as to necessitate widely different heating-surface to oil-volume ratios in centre and wing tanks. An example showing the distribution of heating surface for a typical case is given in the paper.

Under the heading "Cargo Heating Tests", the report describes an investigation into the operation of the heating coil in one of the wing tanks. Measurements were made of steam flow to determine heat input, and of the pressure and temperature gradients along the coil to ascertain the effectiveness of the coil in relation to its length. The extent of waterlogging of the coil was determined, and separate heat-transfer coefficients found for steam and water. Some degree of confirmation was obtained for the coil design procedure put forward in British Ship Research Association Report NS. 34, as can be seen from the Appendix.

INTRODUCTION

Information on heat losses is fundamental to the realistic sizing of heating coils and for the specification of boiler capacity. Although several theoretical studies have been published little effort has been made hitherto to measure the actual heat losses from oil cargoes experimentally. This report deals with measurements carried out on two ships, of 18 000 (Test 1) and 19 000 (Test 2) dwt, carrying typical Venezuelan crudes having the following characteristics:

Test	Oil	Viscosity sec Redwood No. 1 at 100°F	Specific gravity at 100°F
1	Tia Juana Pesado	10 000	0·956
2	Bachequero	6500	0·968

*Assistant Engineer, British Ship Research Association.

INSTRUMENTATION AND PROCEDURE

Heat losses were measured by means of heat flowmeters of a type developed and manufactured by the Technical Physics Department, T.N.O., Delft. This flowmeter consists of a long, helical coil in fine constantan wire wound into a close, flat spiral and copper-plated on one side. The spiral is embedded in P.V.C. to form a flat but flexible disc about 4 inches in diameter and 0·13-in. thick. The coil constitutes a large number of thermocouples in series and a thermo-emf is generated in proportion to the temperature gradient through the disc. Suitable calibration enables measurements to be made of heat flow through any surface to which the disc is attached, and correction can be made for the thermal resistance of the disc itself.

Measurements were confined to one tank, No. 7 starboard wing tank being chosen as typical of all outboard tanks and for convenience in leading cables to the recording instrument in the centrecastle. It was not considered necessary to include a centre tank in the scope of the measurements, as the rate of heat loss from the deck and bottom of a centre tank would

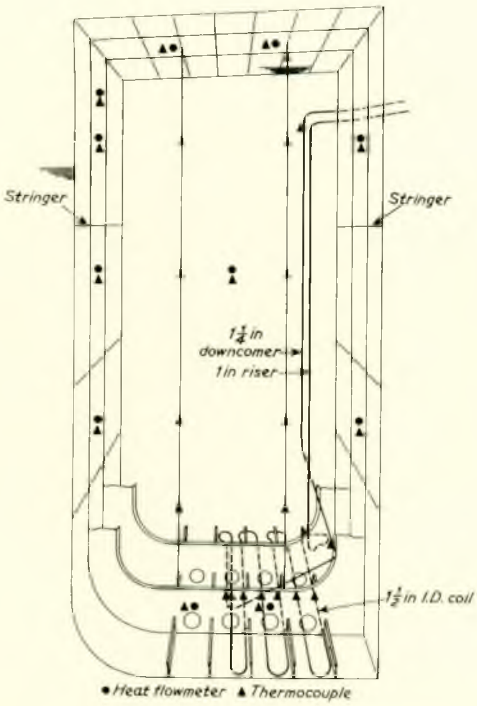

FIG. 1—*Arrangement of instrumentation*

be the same as for a wing tank. Heat flowmeters were fixed by adhesive to the inside surfaces on all four sides, bottom, and deck, in the positions shown in Fig. 1. A thermocouple was peened into the plating alongside each disc for the measurement of the metal temperature.

For the determination of the bulk-oil temperature a number of couples was attached to two wires stretched vertically from top to bottom of the centre bay of the tank. Other couples were used to measure the temperature at the oil surface and in the ullage space, and to determine the temperature gradients in the oil at the tank bottom and side shell.

The leads from the thermocouples and heat-flow discs were brought together and passed through an oiltight seal in the hatch coaming into a terminal box on deck and thence to the ship's office in the centrecastle. The box also contained a vacuum flask, kept filled with melting ice, for the cold junctions of the thermocouples.

During the first trial Nos. 6 and 8 wing tanks were empty and the centre tanks full, while for the second trial No. 7 centre tank was empty as well as Nos. 6 and 8 wings. The presence of empty tanks complicated the trials somewhat in that attention had to be given to heat losses through internal bulkheads, but by this means the test tank could be cooled without cooling the whole cargo and the temperature drop in the test tank was greater than it otherwise would have been.

The trials, which were of 16-17 days duration, consisted of two phases, cooling and heating. Cargo was loaded at about 135°F (57°C) and the contents of No. 7 wing and adjacent tanks were left unheated until the temperature had fallen to about 90°F (32°C). Readings of temperature and heat flow were taken every few hours, and curves were plotted of heat flow against time for the various tank surfaces. From the summation of the areas under these curves, corrected for the thermal resistance of the discs, the total heat loss for the period was obtained. This total was then compared with the heat losses due to the fall in bulk temperature of the oil, as determined by the couples on the vertical wires, weight of oil, etc.

Readings of heat flow and temperature were continued during the heating of the oil and, in the second trial, measurements of heat input were obtained from instrumentation fitted to the heating coil.

RESULTS

The heat losses for the cooling period computed from the heat capacity of the oil and the temperature drop, and the heat losses measured by the heat flowmeters were as follows:

Trial 1

Heat loss:	corresponding to temperature drop	22 427 000 Btu
	measured by flowmeters	20 090 000 Btu
	Difference: Btu	2 337 000
	per cent	11·2

Trial 2

Heat loss:	corresponding to temperature drop	26 580 300 Btu
	measured by flowmeters	21 270 300 Btu
	Difference: Btu	5 310 000
	per cent	25

Two principal reasons are advanced to account for the lower values of heat loss recorded by the heat flowmeters:

a) heat was conducted away by the ship's structure and therefore not measured by the meters;
b) in Test 2 the tank surfaces were sufficiently pitted to give rise to the suspicion that small cavities may have formed under the discs as the adhesive contracted when setting.

The heat losses measured by temperature drop are considered the more accurate. Taking into consideration the possible errors in determining oil bulk-temperature, volume, density, and specific heat, the results should be subject to a maximum error of ten per cent, but are probably a good deal better. Accordingly, the heat flows and transfer coefficients for the flowmeters have been increased by 11·2 per cent for Test 1 and by 25 per cent for Test 2.

In Fig. 2 the heat losses from the principal tank areas, deck, topsides, wetted sides, and bottom, together with the bulk-oil, sea, and air temperatures, are shown for the whole of the measuring period of Test 2. It will be seen that the major part of the heat loss took place at the wetted sides of the tank. As would be expected, heat losses from the deck were subject to wide fluctuations by day and night and according to prevailing weather conditions. Thus the effect of the sun was to heat the deck plating and, if the decks were dry, to reverse the direction of heat flow. Heat losses from the topsides when wet were only slightly lower per square foot than from the wetted sides. The heat lost from the tank bottom was relatively small during the cooling phase, but became more significant during reheating.

The results have been put in the more practical form of heat-transfer coefficients (Btu ft^2 h °F) in Figs. 3 to 9.

Wetted Sides

Fig. 3 shows, for both tests, the heat-transfer coefficients for the wetted sides, based on the oil-sea temperature difference. It will be seen that Test 2 gave higher and more scattered values than Test 1. Test 2 also shows a tendency for the coefficients to rise with increasing oil-sea temperature difference while those for Test 1 remain constant. These differing characteristics can be explained largely by the difference in the viscosities of the two oils and to the prevailing weather conditions.

The thermocouples embedded in the side-shell plating indicated temperatures of from two to six degrees above the sea temperature for Test 1 and from three to eight degrees for Test 2, showing that the major part of the temperature gradient and therefore the thermal resistance was on the oil side of the plate. Couples at the tank side in Test 2, at distances of three, six, and twelve inches from the plating, gave temperatures corresponding to the bulk temperature and no indication of a gradient in this range. Thus the temperature and viscosity gradients were confined to a comparatively thin layer at the

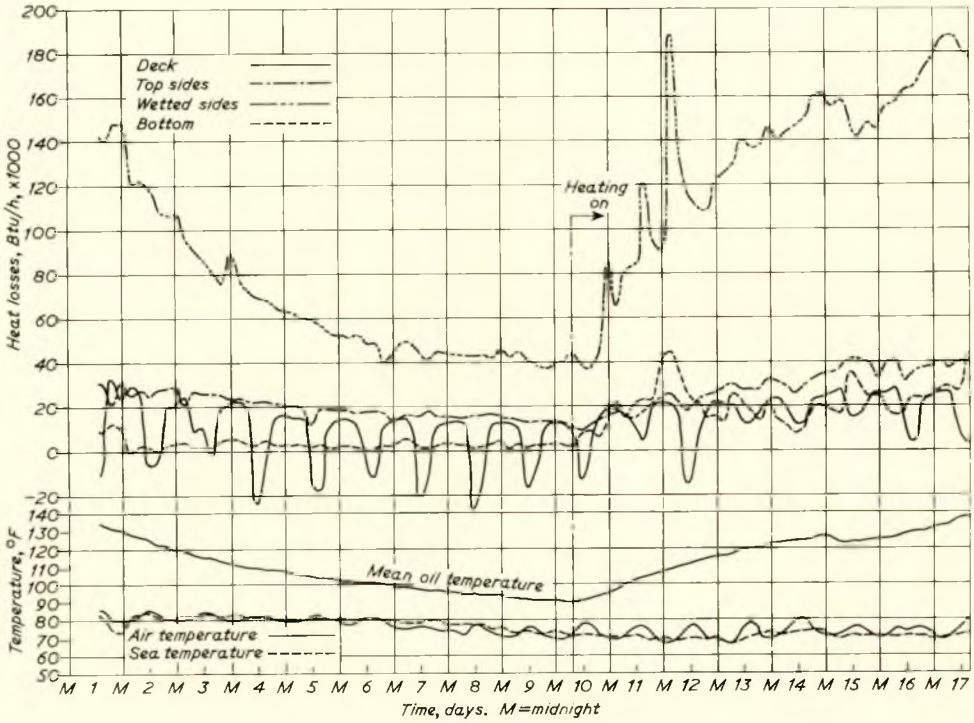

FIG. 2—*Heat losses from wing tank*

metal surface. The effectiveness of this viscous layer as a thermal barrier depends on the viscosity-temperature characteristics of the oil; the more viscous the oil the more sluggish the convection within the film and the greater its resistance to disruption by ship motion. Thus, Test 1, with the more viscous oil and calmer weather conditions, gave lower rates of heat transfer than Test 2 with the less viscous oil and considerable ship motion. The high peak values shown in Figs. 2 and 3 were caused by violent rolling.

With viscous oils some change in heat-transfer coefficient would be expected to take place with changing oil and sea temperature. The effect of raising the oil temperature would be to steepen the temperature gradient at the ship's side, thereby lowering the mean viscosity within the film, and increasing the heat-transfer coefficient. Lowering the sea temperature would also steepen the temperature gradient, but would raise the mean viscosity of the film and thus decrease the coefficient. When the data from Fig. 3 are plotted against bulk-oil and sea temperature these expectations are generally confirmed (see Figs. 4 and 5). Fig. 5 shows a substantial decrease in the coefficient during cooling, when the oil became layered and sluggish, and in Test 2 a marked reversion to the higher values took place when heating was applied, although the tendency is still downward. It would have been an advantage had the tests been continued into colder water, but Fig. 5 strongly suggests that at a given cargo temperature the heat-transfer coefficients would be smaller in colder seas.

Topsides

Some of the foregoing remarks apply also to the topsides. Although external conditions can vary through wide limits, for design purposes we are interested in the worst conditions. In Test 2 the topsides were almost always wet and frequently

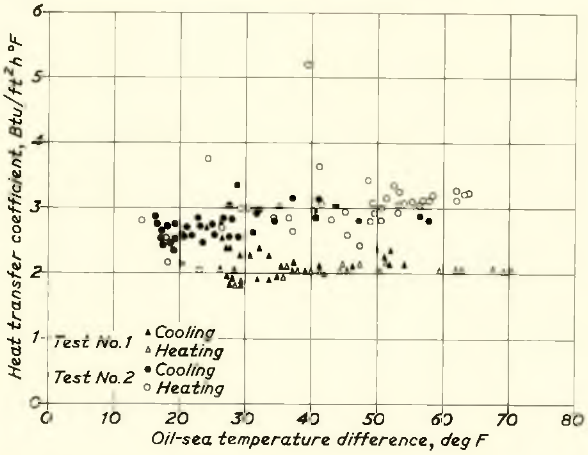

FIG. 3—*Heat-transfer coefficients wetted sides*

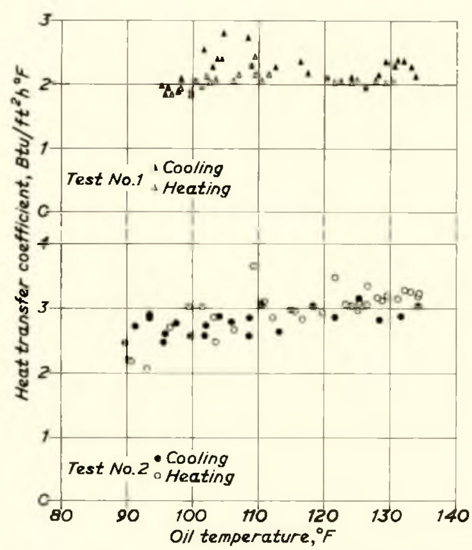

FIG. 4—*Variation of heat-transfer coefficient with oil temperature, wetted sides*

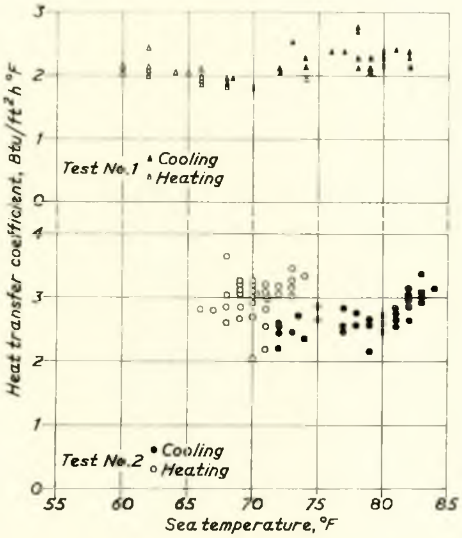

FIG. 5—*Variation of heat-transfer coefficients with sea temperature, wetted sides*

immersed, and Fig. 6 shows that the heat-transfer rate was very little lower than for the wetted sides, so for the purpose of calculating heat losses there may be little point in considering this small area separately from the wetted sides.

Deck

The heat-transfer coefficients for the deck show considerable scatter resulting from the wide range of ambient conditions experienced. Of the positive coefficients shown in Figs. 7 and 8 the higher values correspond to wet decks in dull or dark conditions and the lower to wet decks in strong sunshine.

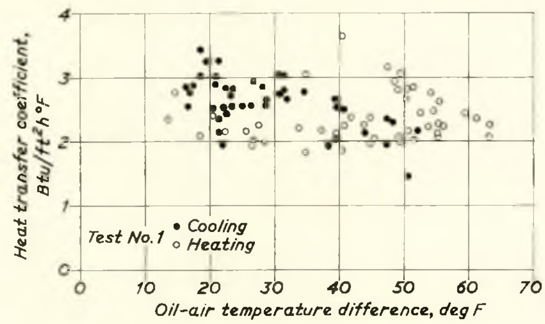

FIG. 6—*Test 2—Heat-transfer coefficients, topsides*

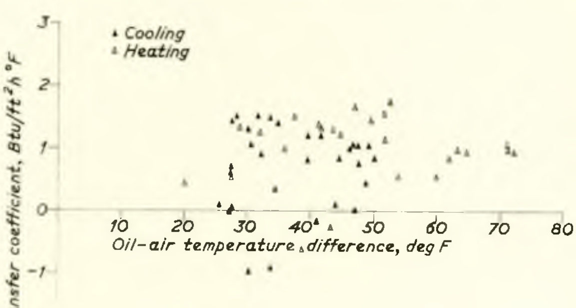

FIG. 7—*Test 1—Heat-transfer coefficients for deck*

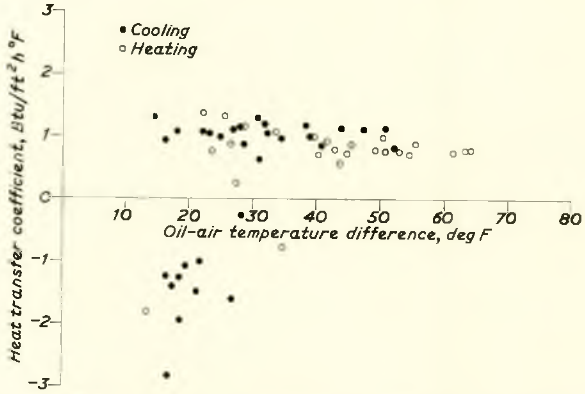

FIG. 8—*Test 2—Heat-transfer coefficients for deck*

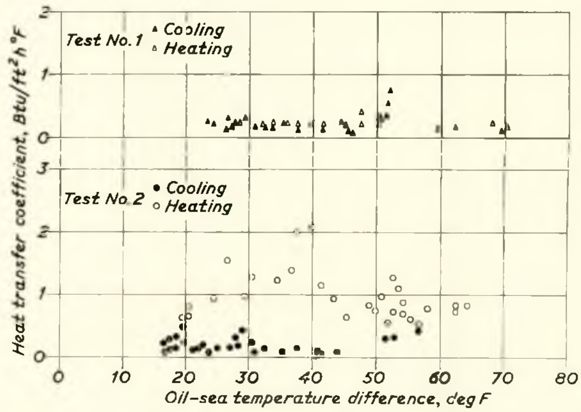

FIG. 9—*Heat-transfer coefficient, bottom*

Negative values correspond to dry decks in strong sunshine. Here again the maximum positive values are of major interest.

Bottom

Fig. 9 shows that during the cooling period the heat loss from the tank bottom was much the same for both tests. Coefficients were very low indicating the presence of a static layer impeding heat flow and resistant to ship motion. Thermocouples at the bottom of the tank showed shell-plate temperatures of only two to five degrees above sea temperature and a steep gradient approaching the interface. During cooling, the oil became stratified so that in fact the temperature gradient extended to the top of the tank.

The application of steam to the heating coils in Test 1 had no effect on heat transfer, whereas in Test 2 appreciably increased heat flows were recorded. The movement induced by heating broke up the strata so that the bulk of the oil was at the mean temperature, with the result that the temperature gradient at the bottom became steeper. Here again the different viscosities probably account for the different behaviour of the two oils. That the heating coils themselves had no influence on this was shown by the heat flowmeters in Test 2, where one meter was in a section close to a coil and the other in a section with no coil, Fig. 1, yet the readings were virtually the same throughout.

The heating coils in the two ships were of the same diameter, approximately the same in surface area, and fixed at the same height of 6 in above the bottom plating.

Laboratory tests with high-viscosity oils have shown that the temperature and velocity gradients associated with horizontal cylindrical heaters are extremely thin and, although there can be turbulence within the film at the steam pressures normally used, the oil outside the boundary layer is at the bulk temperature and is slow-moving. The bottom longitudinals prevent horizontal flow and induce a downward drift of oil, at the bulk temperature, outside the rising column of heated oil. At a comparatively short distance below the heater the oil is undisurbed and heat is transmitted downwards by conduction only.

Longitudinal Bulkhead

For the sake of clarity the heat losses from the internal bulkheads were not shown in Fig. 2, but they are nevertheless of some interest. In the first test the centre tank was full, and No. 7 port and starboard wings and the centre were cooled together. During the cooling period heat flowed continuously from the centre tank to the wing because of the more rapid cooling of the wing tank. The centre-tank temperature also rose much more quickly during heating so that heat flow was again from the centre to the wing.

In Test 2, No. 7 centre tank was empty. With hot oil on four sides the air temperature in the centre tank was very high and the heat loss per square foot from wing to centre tank was small, the direction of heat transfer being reversed when the sun heated the centre tank. Heat losses increased somewhat during the heating process, but were insignificant from a design point of view, particularly as this loading arrangement was untypical. If, however, the centre tank was full and the wing tanks empty the heat-loss coefficient would be as for the transverse bulkhead.

Transverse Bulkheads

Because of radiation to, and the vigorous air movement induced by, the cold shell plates of the empty wing tanks, heat lost from the transverse bulkheads was appreciable, with a mean coefficient of about 0·8 Btu/ft^2 h °F. In view of the large areas involved this heat loss should perhaps be taken into account when sizing heating coils for tanks likely to be isolated, and for end tanks.

SUMMARY OF RESULTS

Because of the differences between the data from the two tests and the scatter of points with varying weather conditions the spread of the results is considerable. However, if the most severe conditions are to be allowed for, the following heat-transfer coefficients might be used:

	Btu/ft^2 h °F
Deck	1·5
Sides	3·3
Bottom	1·5

Note: The designer might consider it more realistic to use average rather than maximum coefficients, in which case he could select suitable values from Figs. 3, 7, 8 and 9.

If losses to empty wing tanks are considered, a value of 0·8 Btu/ft^2 h °F should be used. From measurements taken it is reasonable to assume that the temperature in an empty wing tank will be the mean of the oil and sea temperatures.

Since the first publication of these results data have become available on losses from 3500 sec (100°F) fuel oil in a wing and centre tank of a 50 000 dwt tanker[1]. The following coefficients were determined during acceptance trials in calm weather:

	Btu/ft^2 h °F
Side shell	3·07
Deck	0·70
Bottom	0·48
Tank-wall to empty tank	0·82

The oils used for the B.S.R.A. and T.N.O. tests differed somewhat and this should be borne in mind when making comparisons. It is considered that heat losses from the deck, i.e. from the hot free surface of the oil, would be little affected by viscosity and it will be seen that the average of the B.S.R.A. results (Figs. 7 and 8) would be near the T.N.O. value of 0·7. Viscosity would be expected to have a greater influence on losses from the bottom and side shell, particularly the latter where the stability of the cold insulating layer on the vertical surface would be affected. Thus, the T.N.O. value of 0·48 for the bottom is slightly higher than the B.S.R.A. average, Fig. 9, provided the effects of violent ship motion, B.S.R.A. Test 2, are ignored. A similar comparison of the T.N.O. side shell coefficient and the B.S.R.A. average, Fig. 3, gives the same result.

The T.N.O. report confirms the existence of steep temperature gradients at the tank bottom. Heat losses from and temperature conditions within the bays between longitudinals were the same for all bays irrespective of whether or not a bay contained a heating element. This is useful to know as it is sometimes held that there must be a leg of the heating coil in every bay to obviate cold pockets which would be difficult to drain.

The cargo-heating requirements of a ship naturally depend on the type of service in which the ship is engaged. In many cases it will be sufficient to ensure that the heating system will maintain a given cargo temperature under the most adverse conditions likely to be encountered, while in others it may be advisable to provide, in addition, sufficient reserve to restore the temperature of the oil in a reasonable period should it have been allowed to fall below the desired arrival temperature.

For example, let it be assumed that a viscous oil must be heated from 120°F (49°C) to the minimum arrival temperature of 140°F (60°C) in 48 hours, at sea and air temperatures of 40°F (4°C); specific heat of oil 0·49 at 130°F (54°C); density 59·4 lb/ft^3; oil volumes: wing 21 483 ft^3, centre 43 400 ft^3. The heat losses for wing and centre tanks are shown in Table I, and the heat to raise temperature of the oil from 120° to 140°F (49° to 60°C) in 48 hours:

TABLE I

	Wing tanks				Centre tank			
	Area	Coefficient	Mean $\triangle$T	Btu/h	Area	Coefficient	Mean $\triangle$T	Btu/h
Deck	580	1·5	90	78 300	1060	1·5	90	143 000
Sides	1260	3·3	90	374 000	—	—	—	—
Bottom	565	1·5	90	76 300	1060	1·5	90	143 000
Total				528 600				286 000

Wing 297 300 Btu/h	Centre 590 750 Btu/h

Total heat required:

Wing 825 900 Btu/h	Centre 876 750 Btu/h

Heat input/volume ratio:

Wing 37·8 Btu/h ft^3	Centre 20·2 Btu/h ft^3

If heat losses to empty wing tanks are also taken into account, 26 300 Btu/h would be added to the total for the wing tank and 44 200 Btu/h for the centre tank, for each bulkhead concerned. The totals would then be:

Wing 878 500 Btu/h	Centre 965 150 Btu/h

and the heat input/volume ratios are then:

Wing 40·9 Btu/h ft^3	Centre 22·2 Btu/h ft^3

When oils of lower viscosity than those dealt with here are carried, heat-transfer rates would be relatively higher, but as the transportation temperature would be lower it is expected that the overall losses would be much the same. Similarly, with oils of higher viscosity heat transfer would be relatively low, but this would be counterbalanced by the higher temperature of transportation.

CARGO-HEATING TESTS

Procedure

To supplement the measurements of heat losses from the wing tank in Test 2, instrumentation was fitted to the heating coil in that tank. The coil, shown diagrammatically, in Fig. 1, was of 1½-in bore, 14 gauge aluminium-brass, 181-ft long, with a 1½-in steam downcomer and a 1-in condensate riser of the same material. A liquid-expansion type thermostatic trap was fitted to the coil. Steam was supplied at a nominal maximum pressure of 100 lb/in^2 from a steam-steam generator of 20 000 lb/h capacity. A line diagram of the generator and condensate return system is shown in Fig. 10.

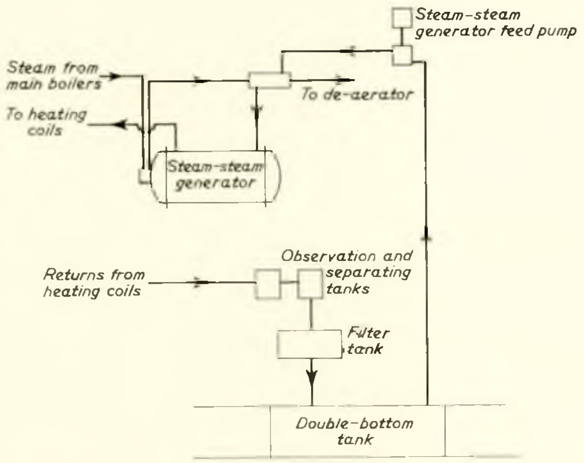

FIG. 10—*Steam-steam generator and condensate return system*

For the trials, a separating-throttling calorimeter was installed on deck at the coil inlet for the determination of the dryness of the steam supply and a three-way cock was inserted in the discharge line, downstream of the trap, so that condensate could be diverted into a measuring tank. A pressure gauge and a thermometer were fitted to measure the conditions just before the steam trap and thermocouples were clamped at intervals along the length of the coil, at the highest point of the tube cross-section, to provide information on conditions within the coil. Ideally the thermocouples should have been peened into the walls of the coil, but as this was not possible they were merely clipped to the coil and this may account, in part, for the low readings given by several couples.

Temperature and Pressure Drop in the Coil

By comparing the coil inlet and discharge pressures, and allowing for the head of condensate in the riser, the pressure drop through the coil was determined. The curve of pressure drop against steam flow in Fig. 11 shows that at the highest rate of flow the drop was seven lb/in^2 or only seven per cent of nominal inlet pressure. As the only pressure loss is due to friction, the drop would be expected to be small; even with high inlet velocity the mean velocity must be comparatively low because of the rapidly diminishing specific volume of the contents of the coil along its length. For practical purposes therefore, condensation can be assumed to take place at constant pressure.

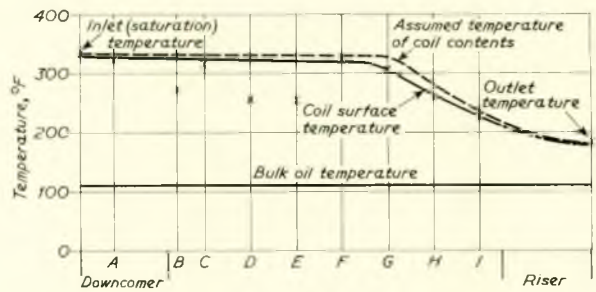

FIG. 12—*Heating-coil surface temperatures*

Fig. 12 is a typical curve of tube temperature (measured at the uppermost point of the tube circumference). The first part of the coil contains condensing steam at saturation temperature, and it is assumed that the resulting condensate will build up progressively as the coil is traversed until, in the region of the inflexion, condensation is complete and the coil is waterlogged. After the inflexion the curve slopes away as heat is removed from the condensate. There will of course be a temperature difference at every section between the coil contents and the tube outer surface. This is indicated in Fig. 12, which is for a high throughput. At lower throughputs the inflexion is less pronounced.

Heat-flow Measurements

Readings obtained during the heating tests are summarized in Table II. Pressures used were in the range 44 to 98 lb/in^2; pressures of 44, 66 and 72 lb/in^2 were obtained by throttling at the inlet valve, the remainder with the inlet fully open. The coil temperature gradients, such as that shown in Fig. 12, were used to obtain separate rates of heat transfer for steam and condensate. The overall heat-transfer coefficients steam-oil are based on the difference between the saturation temperature and the oil bulk temperature, while for the coefficients condensate-oil, the temperature difference was taken as the arithmetic mean of the saturation and discharge temperatures. The surface areas of the downcomer and riser, which were appreciable compared with the area of the coil itself, were included when computing heat transfer.

The principal results from Table II are shown in Figs. 13 and 14. It will be seen that both the heat-transfer rates and the coefficients increase with increasing steam flow. Although heat transfer must obviously increase with increasing

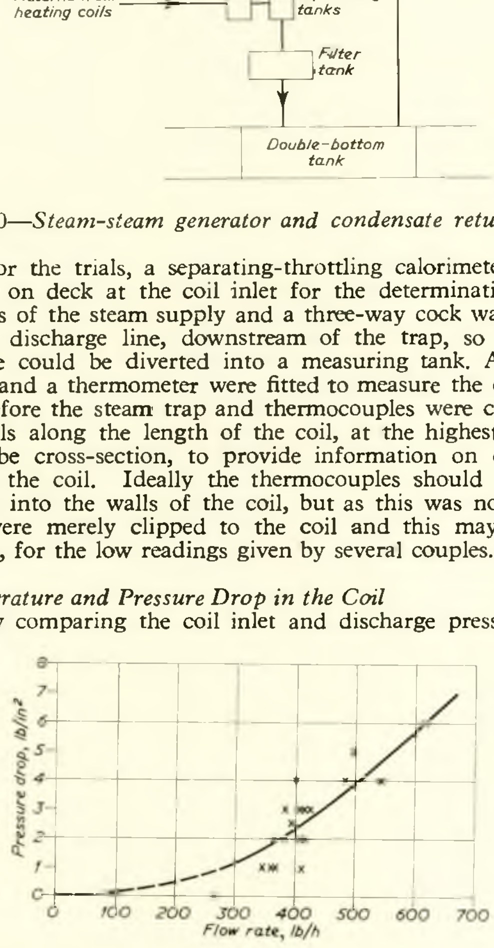

FIG. 11—*Variation of pressure drop with flow rate*

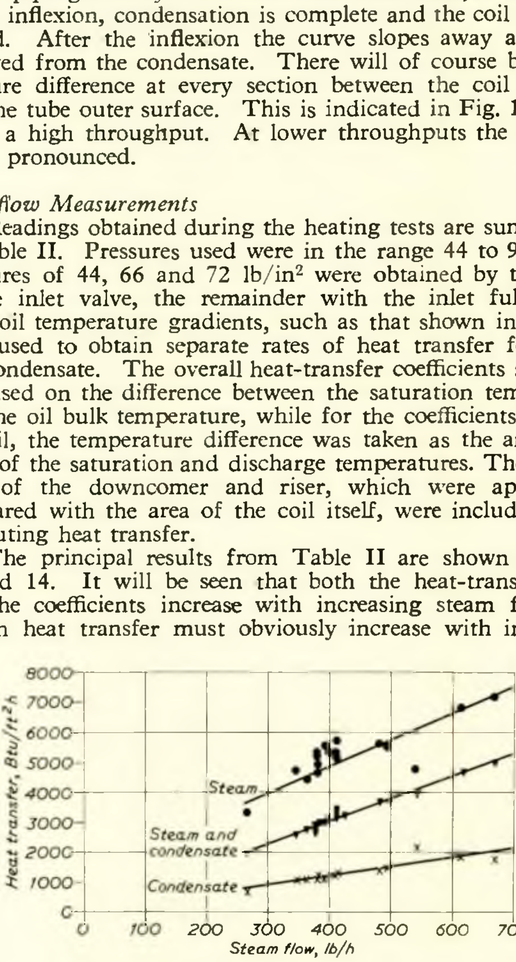

FIG. 13—*Variation of heat transfer with mass flow*

TABLE II

Heating test No.	Steam pressure, lb/in^2	Dryness	Steam flow, lb/h	Steam velocity, ft/s	Condensate discharge temperature, °F	Bulk-oil temperature, °F	Total heat to oil, Btu/h	Steam section: Area, including downcomer, ft^2	Steam section: Heat transfer, Btu/ft^2 h	Steam section: Transfer coefficient, Btu/ft^2 h °F	Condensate section: Area, including riser, ft^2	Condensate section: Heat transfer, Btu/ft^2 h	Condensate section: Transfer coefficient, Btu/ft^2 h °F
1	84	0·89	382	34·4	126	91·5	382 000	65·7	4630	19·7	62·3	1250	9·3
2	84	0·93	411·5	38·9	125	92·3	425 000	57·9	5720	24·4	69·1	1220	9·1
3	86	0·90	414	37·6	130	99·4	414 500	61·6	5380	23·7	65·4	1275	9·8
4	90	0·88	412	34·8	140	101·6	401 500	62·5	5140	22·5	64·5	1240	9·25
5	86	0·89	407	36·4	132	104·2	402 000	–	–	–	–	–	–
6	90	0·90	411	35·4	134	109·4	409 000	61·6	5310	23·9	65·4	1270	10·3
7	44	0·83	266	36·8	122	110·8	248 600	60·7	3340	18·6	66·3	685	7·2
8	66	0·86	365	38·4	130	110·9	348 700	63·4	4440	22·0	63·6	1060	9·6
9	72	0·88	368	37·4	135	112	360 000	–	–	–	–	–	–
10	95	0·92	616	52·1	180	112·2	599 000	73·5	6830	30·8	53·5	1810	12·4
11	98	0·90	400	32·5	138	113	399 400	59·8	5340	23·9	67·2	1200	9·6
12	97	0·90	393	31·7	146	116·9	386 700	55·7	5580	25·5	71·3	1060	8·5
13	97	0·90	382	31·3	145	120·5	379 000	61·6	4950	23·0	65·4	1135	9·5
14	97	0·90	380	31·0	144	123·5	376 000	58·4	5185	24·4	68·6	1070	9·2
15	95	0·89	483	39·8	168	124·8	463 000	67·6	5630	26·6	59·4	1380	10·8
16	96	0·90	670	55·0	196	125·7	626 000	73·5	7220	34·5	53·5	1780	12·7
17	91	0·87	378	31·2	137	123·5	364 000	55·2	5240	25·2	71·8	1040	9·4
18	92	0·88	362	30·1	137	126·6	353 000	–	–	–	–	–	–
19	92	0·87	496	40·9	163	128·1	468 000	68·9	5550	27·2	58·1	1470	12·3
20	93	0·91	425	36·2	164	129·4	415 000	–	–	–	–	–	–
21	90	0·87	541·5	45·0	186	132·1	496 000	87·2	4770	24·1	36·8	2180	17·4
22	93	0·86	347	28·0	143	134·6	330 000	55·2	4740	24·1	71·2	1060	10·3

steam flow, the heat-transfer coefficients for a given steam pressure would not normally be affected significantly by variations in steam velocity. A probable explanation is that the efficiency of heat transfer from the "steam" section, as defined by the inflexion on the temperature gradient, depends on the extent to which condensate accumulates within the section. At high steam velocities the clearance of condensate from the section would be more effective than at low velocities. Thus, although the thermocouples indicate the point where condensation is complete, the actual area effectively condensing steam is unknown.

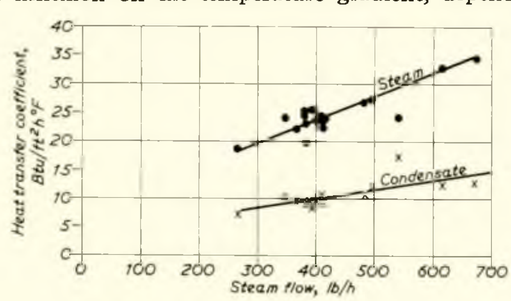

FIG. 14—*Variation of heat-transfer coefficient with steam flow*

At the higher velocities there is good agreement between the experimental coefficients and calculated values based on B.S.R.A. Report NS. 34[(2)], as shown in Fig. 15. This method of calculating heat transfer from horizontal grid coils, derived from turbulent heat-flow theory and supported by laboratory tests, is given in the Appendix.

When the points of termination of condensation are plotted against steam flow it can be seen that the coil was operating at considerably less than the maximum possible throughput, Fig. 16. During the tests the flow rate was varied by adjusting the discharge temperature of the trap; further raising the discharge temperature might have increased the flow to 1000 lb/h or more, although at the expense of economy.

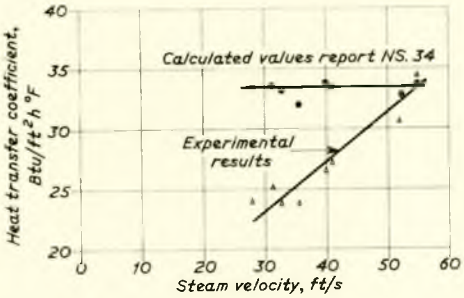

FIG. 15—*Comparison of experimental and calculated coefficients*

In this ship, returns from the heating coils were discharged, after passing through the observation and separating tanks, into a double-bottom storage tank, where a large proportion of the remaining sensible heat was lost. In a system employing thermostatic traps and with no provision for heat recovery in the return circuit, thermal efficiency is clearly dependent on the emission of the greatest possible amount of sensible heat

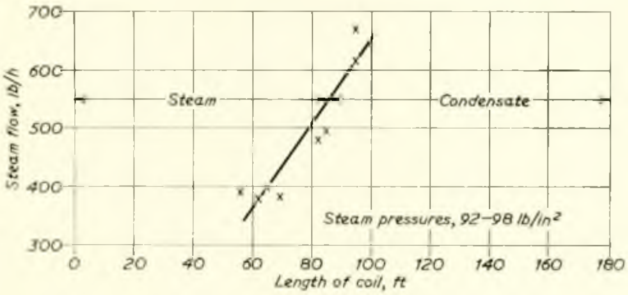

FIG. 16—*Lengths of steam and condensate sections*

from the coils themselves, that is on using a low condensate discharge temperature. When steaming at rates less than the designed maximum, setting the trap to open at only slightly above the oil temperature is practicable, and gives good economy, but in designing for full load a discharge temperature must be chosen that will not result in excessive length of coil. However, the design method for the steam-condensing section of the coil given in the Appendix, and the heat-transfer coefficients for water given in Fig. 14, for the condensate-cooling section, should be of assistance in deciding the relative section lengths to achieve a reasonable compromise between economy of coiling and economy of steam.

Thermodynamic Traps

This type of trap, which passes condensate without regard to temperature, is also commonly used. As discussed in B.S.R.A. Report NS. 34, short coils designed for the emission of latent heat only, in conjunction with these traps, can be used with economy provided that the exhaust system contains means to recover the heat in the returns. For large installations a form of grid is sometimes employed, having several short coils in parallel, connected to downcomer and riser by headers.

ACKNOWLEDGEMENTS

The author acknowledges the permission of the Research Council and Director of Research of the British Ship Research Association to publish this paper. The co-operation and assistance of Shell Tankers (U.K.) Ltd. are gratefully acknowledged.

REFERENCES

1) VAN DER HEEDON, D. J., and MULDER, L. L. "Heat Transfer in Cargo Tanks of a 50 000-dwt Tanker". Studiencentvum T.N.O. voor Scheepsbouw en Navigatie, Delft.
2) COUCHMAN, A. A. J., DOWIE, W. F., and MCCLIMONT, W. "Heating of High-viscosity Oil Cargoes". *Trans.I.Mar.E.*, Vol. 78, p. 53.

APPENDIX

DETERMINATION OF HEATING RATE FROM REPORT NS. 34

The heat-flow equation is:

$$H = 0{\cdot}23\left(\frac{ag\rho ck^2}{v}\right)^{\frac{1}{3}}\theta^{4/3},$$

where:

H = rate of heat transfer, Btu/ft² h,
a = coefficient of volumetric expansion of oil, °F,
g = acceleration due to gravity, taken as $4{\cdot}17 \times 10^8$, ft h²,
ρ = oil density, lb/ft³,
c = specific heat, Btu/lb °F,
k = thermal conductivity, Btu/ft h °F,
v = kinematic viscosity, ft²/h,
θ = temperature difference $(t_s - t_b)$, °F.

This can be expressed as

$$H = (\text{factor } A) \times (\text{factor } B),$$

where $A = 0{\cdot}23\left(\frac{ag\rho ck^2}{v}\right)^{\frac{1}{3}}$

and $B = \theta^{4/3}$.

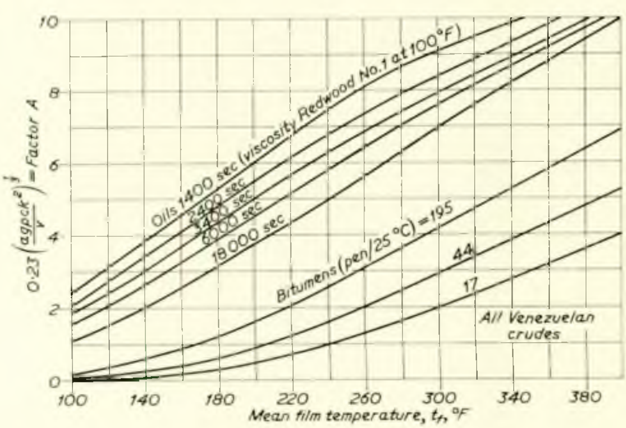

FIG. 17—*Values of factor* A *for five oils and three bitumens*

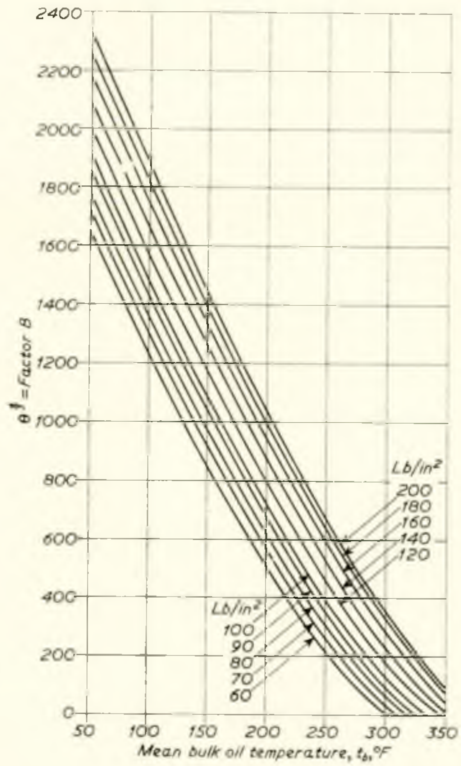

FIG. 18—*Values of factor* B *for a range of steam pressures*

Factor A is given in Fig. 17 for five heavy oils and three bitumens, all based on data for Venezuelan crudes. It should be noted that the base is mean film temperature which is the arithmetic mean of heating-coil temperature and mean bulk-oil temperature. Factor B is plotted in Fig. 18 to a base of mean bulk-oil temperature for a range of steam pressures from 60 to 200 lb/in². The heat-flow rate per ft² of coil surface is therefore computed as follows:

determine the saturation temperature corresponding to the steam pressure in use and consider this to be the coil surface temperature t_s;

decide the oil bulk temperature, t_b, for which the computation is to be made;

then the mean film temperature $t_f = \frac{t_s + t_b}{2}$;

from Fig. 17 read the value of factor A corresponding to t_f;

from Fig. 18 read the value of factor B corresponding to t_b and the steam pressure in use;

then H = factor A × factor B.

CONDENSATE COOLING

As noted in the text of this report, the surface area necessary to extract a given quantity of sensible heat from the condensate can be approximated by using the heat-transfer coefficients for water in Fig. 14.

Written Discussion

MR. J. G. ROBINSON (Member) wrote that this paper should prove very useful to tanker owners and the designers of heating coils. The information on steam consumption for given heat losses should be particularly useful, as in the past ships had not been able to supply sufficient steam. For example on the second of the two ships mentioned, it was the actual steam generating capacity of the steam/steam generator that was the limiting factor, and not the heat transfer capacity of the coils.

The effect of the steam flow rate on the heat transfer coefficient was interesting (see Fig 14). Assuming that the coils must transfer a certain amount of heat an hour, and that the inlet and the outlet temperature were fixed, the quantity of steam used per hour would also be fixed. It followed that if two or more coils were used in parallel, the flow in each would be less and the heat transfer coefficients would be smaller, both for heat and for condensate. It would therefore appear that greater length of coil would be required when coils were used in parallel.

Mr. W. Tipler, M.A. (Member) wrote that anyone who had conducted trials on an operational vessel would appreciate the efforts required to obtain the experimental data presented in this paper.

In view of the undoubted magnitude of these efforts it was regrettable that the maximum use had not been made of the information obtained.

In his introduction the author mentioned theoretical studies of rates of heat loss from oil cargoes, but he neither gave references to these, nor attempted to compare the practical findings with these theoretical predictions, unless reference (2) was one of them. This was a grave omission, since the precision of design work could only be advanced on a sound basis if every opportunity was taken to correlate practical findings with each other and to compare them with theoretical studies. In this way reliable design formulae could be evolved by extracting the maximum information from costly practical investigations.

Perhaps Mr. Saunders could amplify this aspect of the subject in his reply to the discussion.

MR. H. REIK, M.Sc. (Eng.) wrote that the paper was very useful in giving heat loss figures which could be used for anticipating the behaviour of crudes or similar materials during transit conditions.

When designing new tankers it might be worthwhile considering the possibility of insulating the sides of tanks which would be exposed to direct cooling by sea water. This should become more and more important where high pour point crudes would be carried.

The result of insulating sides would be either speedier heating to delivery temperature, or the possibility on ships which did not carry steam to employ other heating methods such as electric surface heating or similar.

With materials like bitumens, which had to be kept at higher temperatures, thermal insulation would be essential and the question of maintaining temperature instead of reheating should be considered seriously.

The economics of thermal insulation and maintaining operating temperatures depended on a number of factors, such as, the lowest temperature at which the material could be carried, the feasibility of fitting thermal insulation and the cost penalty, the possible benefits from not having to provide for a heavy reheating boost at the end of the journey, possibly in a number of storage tanks at the same time.

Generally it could be said that:

a) whenever heat losses during the journey were considerably more than Btu required for reheating, boosting at the end of the journey would be preferable;

b) whenever heat losses during the journey were less than Btu for reheating, temperatures could usefully be maintained at a higher level;

c) where the material itself must not fall below a minimum temperature, a slightly higher temperature might be chosen and maintained, thus reducing the boosting required at the end of the journey.

The author's comments on these points would be welcome.

Author's Reply

In reply to the contributors, Mr. Saunders wrote that many variables were involved in the heating process but in a grid coil/steam trap system of the type investigated with a given length and operating at given oil and steam temperatures, the heat transfer per square foot was controlled by the steamtrap setting. For instance, raising the condensate discharge temperature reduced the amount of heat which had to be removed per pound of condensate and thus the length of coil devoted to cooling condensate, was reduced, leaving more space for condensing steam. Provided it were available, steam would enter the coil at a greater rate, and again, provided the steam-trap capacity were not exceeded, the coil would continue to operate at an increased rate of heat transfer. Heat transfer, i.e. coil operating efficiency, was improved by the increased velocities involved although overall thermal efficiency might be reduced through non-utilization of sensible heat.

With regard to Mr. Robinson's question, it was not easy to see how this general picture would be changed if the coil were split into a number of parallel branches of the same total length. There was no obvious reason why the branched coil should not be capable of an equal throughput of steam, and thus the same heat transfer per square foot though efficiency might suffer somewhat because of the reduced rates of flow in the branches.

The advantages of the branched coil were realized in the splitting up of very long coils, e.g. 4×200 instead of 800 ft and possibly in the achievement of better heat distribution over the bottom of a tank. Perhaps too much importance should not be attached to the latter point. Fig. 1 showed that the coil covered barely half of the tank bottom, and furthermore, though a large part of the coil in the tests contained only hot water, no difficulties were experienced in heating and draining the tank.

For Mr. Tipler, and anyone else interested in the application of heat transfer theory to viscous oils, a bibliography was given.

The physical properties of the test oil were also given.

The author would have done more in the way of comparing the full-scale test data with theory had he thought that the theory was used. In the absence of practical data most designers had merely employed factors relating ft^2 of heating surface to the oil volume, derived from long experience. The object of the work of the British Ship Research Association was to find a better basis for designing these costly installations than rule of thumb.

The author agreed with Mr. Reik that thermal insulation would be a good thing, if a material could be found which was cheap enough and easy to install and protect from deterioration. He wondered what form of insulation Mr. Reik had in mind, and when the latter spoke of electrical surface heating, what form this would take, how the heaters would be disposed and how he would deal with the electrical insulation problem.

Mr. Reik's final points (a), (b) and (c) were valid, whether or not thermal insulation was fitted. It was possible, taking into consideration the type of oil, the probable weather conditions, the duration of the voyage, and so on, to decide in advance the optimum heating pattern for the voyage. This was already done to some extent, but a simple form of calculation could no doubt be developed.

BIBLIOGRAPHY

Calcul du Rechauffage a la Vapeur des Soutes a Mazout (Calculations of the Heating of Fuel Oil Tanks by Steam) by G. CAHEN, and P. BAIN, Bull. Ass. tech. marine, 1951, No. 50, p. 91

Die Wirtschaftliche Heizung der Tankerladung (The Economical Heating of Tanker Cargoes) by N. GASPAROVIC; Hansen, April 1960, Vol. 97, p. 839.

Heizungsanlagen auf Tankschiffen (Cargo Heating Installation Tankers) by F. ZIMMERMAN, Hansa, February 1960, Vol. 94, p. 393.

Lastvörmming I Tankfartyg. Ett teoretiskt bidrag till fragon om storleksberoendet (Cargo Heating in Oil Tankers. A theoretical Contribution to the Influence of Ship Size upon Heat Requirements) by L. NORDSTROM; Swedish Shipbuilding Research Foundation Report No. 26, 1961.

OIL PROPERTIES
(Test No. 2)

Viscosity	°F	cS
	80	3840
	120	680
	150	260
	200	39
	250	35

Density: 0·9813 g/ml at 40°F; 0·9100 g/ml at 250°F (straight line)

Specific heat: 0·48 at 100°F; 0·55 at 200°F (straight line)

Thermal conductivity

$$K = 7{\cdot}47 \times 10^{-2} - 2{\cdot}227 \times 10^{-5} T$$

Btu/ft-h °F