TABLE 1 PREDICTED AND TEST PERFORMANCE FOR INSULATION

	Predicted	Test
Heat leak from hot wall (average over test section) q_h Btu/(hr-ft² of pipe wall)	382	383
Heat leak to ventilating air (average over test section) q_v Btu/(hr-ft² of pipe wall)	362	376
Heat leak to machinery space (average over test section) q_∞ Btu/(hr-ft² of pipe wall)	20	11.3
Outside-surface temperature, above room temperature (average over test section) $T_o - T_\infty$, deg F	4.5	3.0
Cooling-air temperature rise (10.5-ft test section flow length) $T_{f\text{out}} - T_{f\text{in}}$, deg F	25	25.3

magnitude is in error for the main reason that the magnitude of R_o was based upon the manufacturer's figure of a heat loss for the asbestocel of 62 Btu/(hr-ft²-100 deg F temperature difference) where the temperature difference is measured from the surface being insulated to the room condition. Undoubtedly, this figure is based on a temperature difference on the order of 100 to 200 F. In the tests, however, the corresponding temperature difference amounted to only 13 deg F (avg). As a consequence, radiation and convection transfer coefficients within the cells of the material and from the outside surface were probably lower, and the effective resistance R_o greater than used in the prediction. Another factor contributing in minor degree to the difference between predicted and actual q_∞ is that the predicted value of R_r was based upon a wall emissivity of 0.9. With the aluminum surfaces actually used, a much lower emissivity and higher R_r existed. However, since $R_f << R_r$ for the system and mass velocity tested, the influence of R_r was minor.

This structure has an over-all thickness of $4^3/_4$ in. However, by using a primary-insulation-layer material with $k = 0.0467$ Btu/(hr-ft²-deg F/ft), an amosite asbestos felt of approximately 10 lb per cu ft density, the thickness of the structure can be reduced to $3^5/_8$ in., a saving of $2^1/_4$ in. on the diameter, for the same performance.

Summary and Conclusions

1 A ventilated thermal-insulation structure can be made for insulation of high-temperature surfaces which will have the characteristics of small over-all thickness, low heat leak to the machinery space, and high heat leak from the working substance, when compared with the conventional types of insulation.

2 For a gas-turbine plant, the high heat leak from the working substance may not be excessive. The blower-power requirement and size are small. The number and size of air ducts are not excessive.

3 A method for predicting the performance of the ventilated structure has been presented, and the predictions have been confirmed by the limited experimental data so far obtained.

Acknowledgments

The work reported in this paper was made possible by the U. S. Navy Office of Naval Research and the Bureau of Ships. The authors are appreciative of the assistance provided by Mr. Jack Reed, graduate student, and Mr. H. J. Jesperson, research associate, in preparation of calculations and design of the test system.

Discussion

E. A. Richardson[6] and G. A. Richardson.[7] The theoretical and experimental investigation of the ventilated form of insulated enclosure deals with one possible form of control of heat loss from the working substance, heat limitation to the outside of the enclosure, and protection of structural members. In the proposal, a pipe carrying gas at 1300 F is insulated on the outside by primary insulation, while cooling gas is circulated in the passage formed between the primary, and the secondary insulation. It is noted that the rise in cooling-fluid temperature is about 60 deg F in the example given and that the heat so picked up is wasted.

The concept of effective thermal isolation is not new, in fact the partners investigated means for securing equivalent results nearly 15 years ago. The insulation of boilers in steamer holds to prevent excessive heat leakage and cooling requirements was recognized as desirable. It was seen that progress of all high-temperature high-pressure processes would depend upon thermal isolation combined with recovery of heat lost from the working fluid.

One example of such an investigation was the insulation of the walls of a petroleum cracking chamber operating at 850 F and under 750 psi pressure. A slightly permeable brickwork was conceived placed inside of, and slightly spaced from the shell, while fluid was pumped into this space at such rate as to secure substantially atmospheric temperature in the shell. Using oil to be cracked, it was possible to pump 1 or 2 per cent of the process oil through the permeable layer, whereupon the "cooling" fluid would be heated to process temperature, recovering virtually all of the heat normally lost. As the steel wall no longer involved creep behavior and no longer had a temperature gradient, it became possible to use about one-half as much flanging-grade steel in the shell as of the somewhat alloyed and stainless-clad steel actually used, while securing a much safer design.

It becomes of interest to see how the partners might propose to handle the case discussed by the authors of this paper. But first, it must be realized that the partners had removed the temperature limit to cracking processes in so far as structural materials might be concerned and imposed temperature limits, if any, on the basis of refractory behavior.

Retaining the "Superex" and asbestos-sponge felt shown in Fig. 9 of the paper, the partners might substitute $^1/_2$ in. of glass wool for the asbestocel layer and remove the aluminum shell immediately within this layer. They would then cause air to be passed radially inward through the glass-wool layer into the ventilating space, the air to be exhausted therefrom as at present. Suppose the entering-air temperature is 80 F while the outer surface of the glass wool is at 90 F. The heat loss from the outer surface cannot exceed 5 Btu per sq ft per hr for this 10 deg F drop, for, obviously, convection cannot occur.

Having set the outer-surface temperature 10 deg F above the entering-air temperature, we can assume a set of values of hot-surface temperature of glass wool, and determine therefrom the value of nC_pL/k_s, where n is the rate of fluid permeation in pound-mols per square foot per hour, C_p is the molal specific heat in Btu per 1 deg F per lb-mol, L is the insulation thickness (normally in feet, but for this case in inches), and k_s is the effective conductivity of the glass-wool and air combination. It is then possible to tabulate the value of hot-surface temperature as T_h, the other quantities given, and add data on the heat to the air and the required temperature of the outer surface of the primary insulation to supply this heat by radiation to the glass wool, convection being neglected.

TABULATION FOR FIRST SUBSTITUTION

T_h	nC_pL/k_s	C_p	k_s	n	q	T_{pi}
500	3.76	6.91	0.480	0.523	1528	738
400	3.47	6.88	0.444	0.447	985	614
300	3.10	6.85	0.413	0.373	563	479
200	2.49	6.83	0.386	0.281	230	320

Note: Term k_s is in Btu per hr per 1 deg F per in.; q is in Btu per sq ft per hr; T_{pi} is temperature, deg F, outer surface of primary insulation.

If the heat leak from the hot wall is of the order of $382 = q_h$, then T_h might be 245 F by interpolation, $T_{pi} = 390$ F, and 0.32

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lb-mol of air per sq ft per hr are required. This is 9.25 lb per sq ft per hr. For the 15-ft length of pipe, and the annular space as given, this corresponds to $G = 1.08$ lb per sq ft per sec, or about one half of that required by the authors. However, the air is now heated from 80, to 245 F, or 165 F, a value much larger than the 60 F in one example, or the 25 F in the test case. A decided reduction in cooling air has occurred. Furthermore, the heat loss from the outer surface is approximately constant and negligible, and not rather widely varying lengthwise as in the authors' case. Our method does not permit convection from the outer surface.

Actually, we should feel that such a method of treatment is makeshift in character. We should prefer to place a primary layer of permeable insulation inside of the pipe, as well as the annular space and the secondary insulation. In such case, the cooling air would be bled from the compressor, dividing it into two parts. One part would be caused to enter the annular passage, after a slight pressure boost, at a temperature of the order of 400 F, while another part would be cooled to room temperature for passage through the secondary insulation. A slight pressure boost of the bled air is required to insure flow at a proper rate through the permeable layers. The whole of the bled air, after it has served its purpose as a coolant, mixes with the hot gases passing to the turbines. The effect is that radiation has been almost entirely eliminated at the cost of a slight boost in pressure of the bled air, while the temperature limit of the cycle, in so far as piping is concerned, has been eliminated. It would be just as practicable to use gases at 3000 F as at 1300 F. The gases not bled are at a slightly higher temperature, leaving the combustor, than would be normal. The pipe is at atmospheric temperature. Although larger in diameter, it can be lighter in weight and safer than the stainless-steel piping shown by the authors, as well as much less costly.

Assume the value of k_s (per ft, not per in.) as 1.0, the mean specific heat of air between 425 and 1300 as 6.70, and an outer temperature of primary layer 3 per cent of the difference of 875 F (26 F) above the entering air temperature, the value of nC_pL/k_s becomes 3.50, so $nL = 0.52$ (L now in ft). Using $L = 0.25$ (3 in.), then $n = 2.08$ lb-mol per hr per sq ft of 400 F bled air. Since the $^1/_2$-in. glass-wool layer requires (see table in foregoing discussion) about $n = 0.45$ lb-mol per hr per sq ft, this is the amount of bled air which must be cooled from 400, to 80 F. Thus 0.45 mol cool air, 1.63 mol hot air are bled per hr per sq ft. The heat removed from the bled air (0.45 mol) is 1100 Btu per hr per sq ft. The total heat removed from the hot gases is 18,300 Btu per hr per sq ft, fully returned in the "cooling" air. However, these results are excessive, in that it is assumed the hot gases can deliver 18,300 Btu per hr per sq ft. Actually, the inside of the primary insulation will be cooler, hence correspondingly less air will be required.

Figures used herein were derived by the partners, partly theoretically, partly by test. The rapid variation of k_s for glass wool tends to confirm the difference in heat leak to the machinery space (20 predicted, 11.3 on test) found by the authors. However, knowing the rapidity with which radiation effects increase with temperature for wool and felted insulations, we would question the statement regarding the saving in primary-insulation thickness through the use of amosite asbestos felt of 10 lb per cu ft density. Actual measurements of conductivity are required in the region above 1000 F, since extrapolation can be hazardous.

We think that it will be obvious that more promising methods of thermal isolation are available than the ventilated type of insulation investigated by the authors. The partners have done much work investigating the use of "permeable plates with fluid permeation between faces" for insulation, cooling, heat transfer, and the like, for a wide range of applications.

Authors' Closure

In considering means of controlling the heat loss from high-temperature surfaces the possibilities of using the thermal-energy storage capacity associated with a mass transfer through a porous material, as proposed by the discussers, should certainly be studied. The idea has been proved in many practical instances. The ordinary kitchen watercooler jug allows water to seep through the porous surface to the outside, where the evaporation provides a substantial cooling in dry climates. The combustion-chamber liner of the aircraft turbojet engine, such as the German Jumo 003, is provided with a large number of small vents through which cooling air is introduced, sweeping the inside of the liner with a blanket of cool air, and effectively isolating the low-carbon-steel outer casing (of sheet metal) from the high-temperature zone. It has been demonstrated that cooling of a rocket nozzle by the passage of a gas or a liquid through a porous material into the nozzle stream has attractive possiblities for maintaining safe temperatures in the nozzle wall.

The possibility of placing stressed parts of the machinery, such as the hot gas-duct wall, on the cold side of the insulating material is recommended by Messrs. Richardson. The possibility of doing this was mentioned in the paper, and should certainly be studied for any specific insulation application.

Certain practical difficulties are apparent in attempting to apply the specific recommendations of the discussers. In placing the primary insulation layer on the hot side of the duct wall, it still will be necessary to use a stainless liner material to hold the insulation material, which is usually of very fragile material, in position; otherwise damage might occur to any turbine machinery which might follow (especially if a ceramic material were used for the insulation). Also, mechanical protection of the cold side of the insulation, adjacent to the cooling-air annulus, would be necessary. The mechanical structure would become somewhat complicated.

It is to be noted that the stainless-steel pipe shown in Fig. 9 of the paper was not intended to represent a hot gas duct; it represented, rather, a means of securing a hot surface for the test setup which would maintain its shape and not scale excessively at the high temperatures tested.

The suggestion for using a $^1/_2$-in. layer of glass wool for the outside insulation blanket (in the machinery space) also meets with the objection of mechanical vulnerability. It would have to be supported with perforated metal both inside and outside. A dirty atmosphere also might seriously impair the porosity; this may be a valid objection on shipboard, even though the atmosphere is relatively quite clean. The thermal isolation possible with the discussers' proposal is obviously greater than that possible with the ventilated arrangement discussed in the paper.

It is not clear from the discussers' comments what the significance of the term nC_pL/k may be, nor how it is to be determined or utilized. By considering an energy balance at a point on the duct surface at thermal equilibrium

$$\underbrace{n \cdot C_p \cdot \Delta t \cdot \delta A}_{\substack{\text{associated with}\\ \text{cooling air mass}\\ \text{transfer}}} = \underbrace{U \cdot \delta A \cdot \Delta t}_{\substack{\text{associated with}\\ \text{conduction through}\\ \text{insulation material}}}$$

If the convection resistances at the two surfaces of the insulation are neglected, this expression reduces to

$$nC_p = \frac{k}{L}$$

and it would appear that the value of nC_pL/k should be about 1.0, rather than the value of 3.50 given. Of course much depends upon the significance of k when there is a flow of air through the insulating material.