EXPERIMENTAL COAL-BURNING GAS TURBINE

Exhaust-Heated Cycle

J. W. STACHIEWICZ AND D. L. MORDELL

DEPARTMENT OF MINES AND TECHNICAL SURVEYS-OTTAWA

MINES BRANCH No. 867

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PREFACE

The work described in this report has a unique significance. It was the first time that the initial research and development of a completely new power plant was carried out in Canada and perhaps peculiarly appropriate that it should result from co-operation between McGill University and the Department of Mines and Technical Surveys of the federal government.

The report describes only what might be regarded as a first phase. It was primarily a research effort designed to reveal and elucidate the problems that will be encountered in connection with the design and operation of a prototype power plant, which could be regarded as a second phase. At the same time, as is always true in engineering, a considerable amount of development was involved, and by the end of the test program practical solutions had been worked out for many of the problems encountered. Many papers have been presented before technical societies and published in technical journals throughout the world describing particular stages of the work, but the present report is an account of the whole program, from inception to conclusion, and as such should have a unique value. As to the value of the work itself, quite apart from its contribution to knowledge in its particular field, the conduct of this work in a university laboratory over a 7-year period contributed much to the training of many engineers who are now in Canadian industry and Canadian universities. It is not without interest that two direct applications of the work done have arisen in the design of steam generators for nuclear-power plants and in the design of furnaces to make a useful product out of rice hulls.

Although the writer had the privilege of directing this work throughout its duration, its successful execution was the result of a team effort and many people contributed in various ways. The hardest step was perhaps the actual initiation of the program. The late W. A. Newman, formerly Director of Research, Canadian Pacific Railway, and Dr. T. E. Warren, formerly of the Mines Branch, Department of Mines and Technical Surveys, gave invaluable help. Since 1954 E. R. Mitchell, Senior Combustion Engineer, acted as coordinator for the Department. His invaluable help and technical assistance in the preparation of this report is warmly appreciated. In the original planning and design Dr. J. T. Rogers and Professor R. E. Chant were invaluable colleagues. The early experimental work in Ottawa was carried out by H. P. Hudson, of the Mines Branch, who also advised on the initial furnace design. The responsibility for the actual construction of the experimental set up and for supervision of the test program lay on R. W. Foster-Pegg, who was also responsible for the development of the furnace and coal-handling system and who was ably assisted by A. W. Haddon. Throughout the tests the best instrumentation facilities were enjoyed and were arranged for by J. J. Gravel. Our indebtedness to the Locomotive Development Committee and J. I. Yellott is acknowledged in the text. The Dominion Steel and Coal Corporation Limited supplied all the coal consumed.

The senior officials of the Department of Mines and Technical Surveys and of McGill University at all times went out of their way to facilitate the joint operation. In the metallurgical examination in particular, many members of the Department gave much help. Finally, but not less warmly, the writer would like to express his thanks to Professor J. W. Stachiewicz, who, apart from the invaluable contribution he made on heat-exchanger problems during the work, has produced this comprehensive account of the whole program.

D. L. Mordell
Dean of the Faculty of Engineering
McGill University

PART I

PLANNING AND DESIGN

INTRODUCTION

The years immediately following World War II were characterized in Canada by a tremendous growth of the economy and an unprecedented expansion of natural resources. The increase in fuel requirements, however, was limited mainly to liquid fuels, in spite of the large reserves of coal available. Not only did fuel oil capture most of the new markets, but it began to displace coal in areas of power generation where, until then, solid fuels had been used almost exclusively.

Thus, gradually, a situation developed where a major proportion of Canada's power requirements (chiefly in the motive-power field) were dependent almost exclusively on oil as fuel.

That this trend could, if continued, result in virtual extinction of a major Canadian industry was a distressing, but obvious, conclusion.

That it could result in a critical shortage of fuel for domestic needs in times of war, when oil would be monopolized by the armed forces, was a fact which could not be ignored.

Thus, when in 1949 a proposal was made by D. L. Mordell[1] to develop a new type of engine to burn low-grade coal as fuel, the Department of Mines and Technical Surveys agreed to sponsor the project.

[1] D. L. Mordell of McGill University, Montreal, at that time professor of mechanical engineering and now Dean of the Faculty of Engineering.

Mordell's proposal (1) (2)[2] was that a gas-turbine engine be adapted to burn coal by means of a thermodynamic cycle which has since become known as the Mordell cycle, or the exhaust-heated cycle.

An agreement was drawn up in 1950 between the Department and McGill University for the design and construction of a test facility for the investigation of problems connected with the development of this type of engine.

• Gas-turbine Cycles

The gas-turbine engine was at that time a relative newcomer to the power-generation field, being used almost exclusively for aircraft propulsion during World War II and burning oil as fuel.

The first successful non-military unit was constructed by Brown-Boveri in Switzerland as a standby power plant for the city of Neuchatel in 1938, and a unit was placed in a locomotive in 1941. By 1949 the oil-burning gas turbine was a firmly established prime mover.

Brown-Boveri were the first to realize that the potentialities of the gas turbine would be greatly increased if it could be made to burn coal, but since in Switzerland oil was the cheaper fuel, this line of investigation was not pursued, particularly as early experiments revealed many difficult problems.

The combustion of coal in a gas turbine presented two main problems:

(*a*) that of developing a suitable furnace to burn the coal efficiently

(*b*) that of ensuring that the fluid passing through the turbine was free of elements which would cause corrosion or otherwise harm the engine.

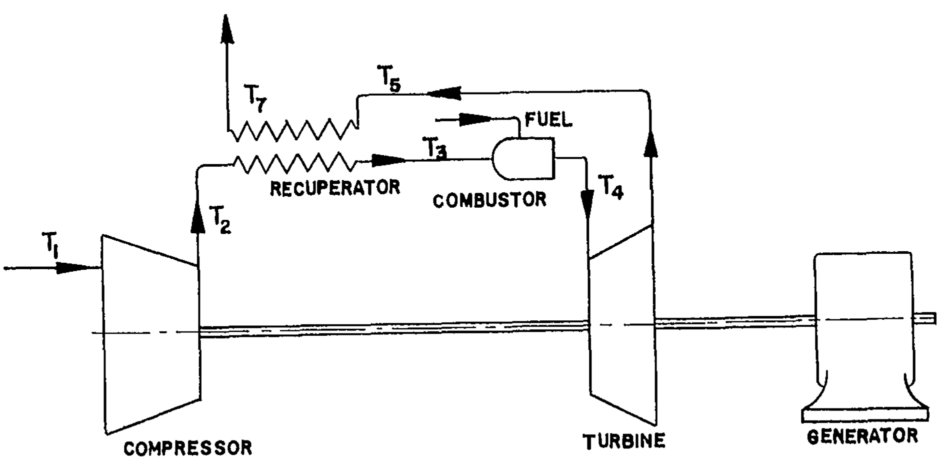

Figure 1. Simple gas-turbine cycle with recuperator.

There are a number of ways in which a gas-turbine cycle can be arranged, depending on consideration of power per unit weight, thermal efficiency, type of fuel, etc.

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[2] Numbers refer to Bibliography.

Figure 1 represents the conventional oil-burning gas-turbine cycle with recuperation. Fuel is burnt in the combustion chamber in the presence of compressed air, and the resulting gases expand in the turbine, developing enough power to drive the compressor and to deliver useful power to the load. Part of the sensible heat present in the exhaust is recovered in the heat exchanger, thus increasing the thermal efficiency of the cycle.

This cycle can be used if coal rather than oil is burnt in the furnace, but then the products of combustion will contain solid particles of ash, slag or unburnt coal, which must be removed in a suitable separator before the gases are allowed to enter the turbine (Figure 2).

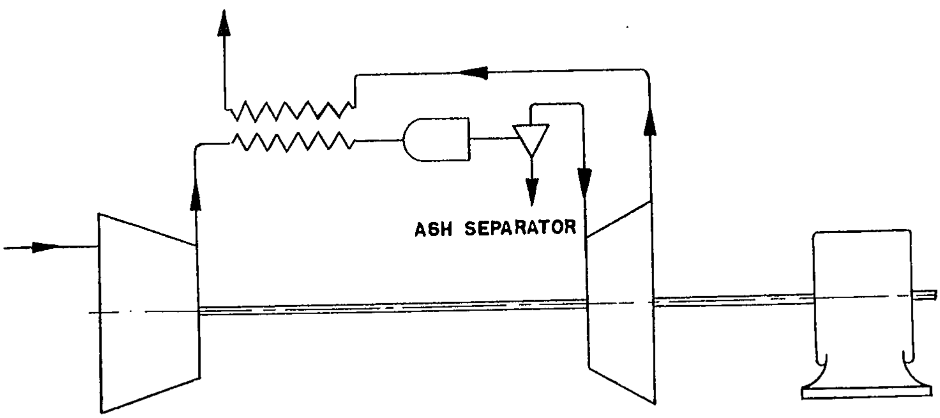

Figure 2. Simple coal-burning turbine with ash separator.

The problem then becomes one of developing an efficient method of removal of fly ash from the gas stream. Although up to 99 per cent of all solid particles can be removed by means of cyclone separators, this can be done only at the expense of a considerable pressure loss. This pressure loss reduces the pressure at the inlet to the turbine and causes a reduction in the turbine expansion ratio which results in a serious diminution in the net power output (Figure 3). Since considerations of turbine-blade life require the removal of more than 95 per cent of the ash particles, the design of a reliable and efficient separator with an acceptable pressure drop has proved a formidable problem.

A very considerable amount of work has been done along this line since 1945 by the Locomotive Development Committee of the Bituminous Coal Research Inc., U.S.A. (B.C.R.) under the direction of J. I. Yellott; but although several million dollars have been expended and an efficient pulverized-coal furnace has been developed, the turbine-corrosion problems have still not been completely overcome (3) (28c).

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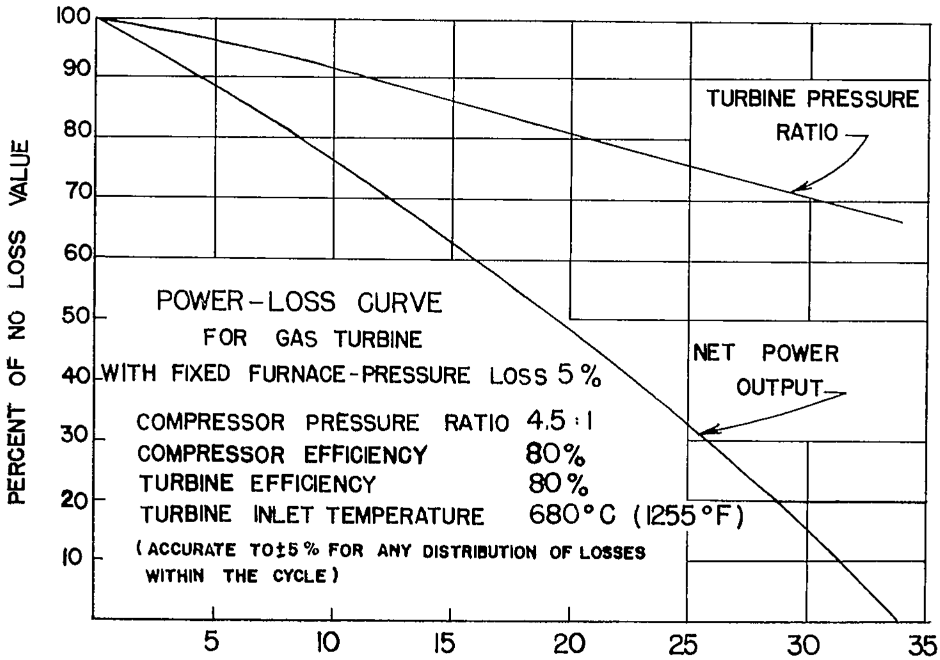

Figure 3. Effect of cycle-pressure losses on power output.

Other lines of investigation include work on the so-called closed-cycle turbine (Figure 4) by Escher Wyss in Switzerland. Here coal is burnt in an external circuit and heat is transferred to the gas-turbine fluid in a heat exchanger. The main advantages of this cycle are:

(*a*) The turbine works on clean air and thus no corrosion problems exist.

(*b*) The air may be used at high pressure, thus reducing considerably the size of all components.

(*c*) The thermal efficiency, particularly at part loads, is high.

The main disadvantages are:

(*a*) high initial capital cost

(*b*) large cooling-water requirements.

• Exhaust-heated Cycle

Mordell's proposal combined the best features of the closed cycle (i.e. circulation of clean air through the turbine) with the relative simplicity of a conventional open cycle.

The cycle is shown in Figure 5.

Atmospheric air is compressed in a compressor and then passes through a heat exchanger, where it is heated to a temperature of about 1,350°F. It is then expanded in the turbine, which drives the compressor and supplies useful power. From the turbine the air is admitted to the furnace, where coal is burnt, and the resulting combustion products pass through the heat exchanger, in which they give up heat to the compressed air.

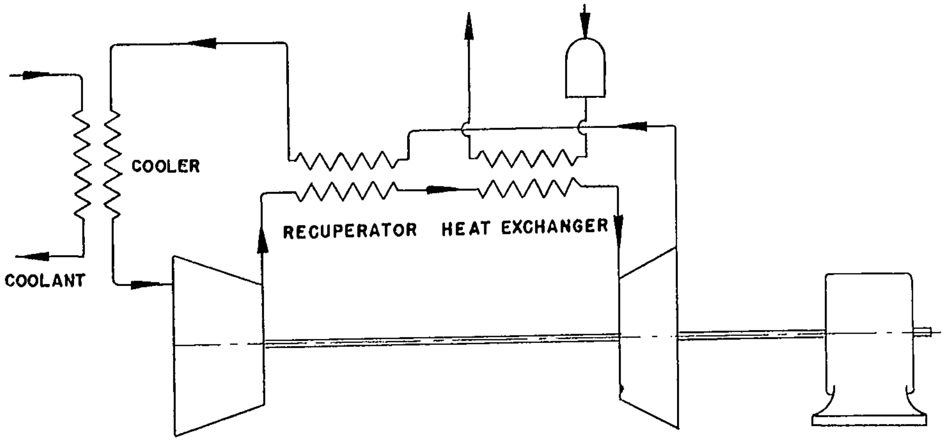

Figure 4. Closed gas-turbine cycle.

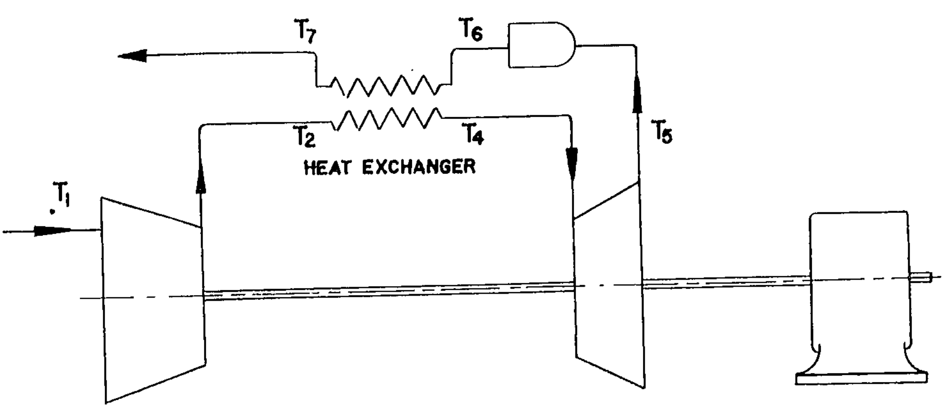

Figure 5. Simple exhaust-heated cycle.

Since the thermal ratio[3] of the heat exchanger must be less than 100 per cent (unless infinite heat-transfer surface is provided), the exchanger air-outlet

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[3] Thermal ratio R_{th} is defined as the actual rise in temperature of the air stream divided by the maximum possible rise, i.e.

$$R_{th} = \frac{T_{\text{turbine inlet}} - T_{\text{compressor outlet}}}{T_{\text{combustion chamber}} - T_{\text{compressor outlet}}} = \frac{T_4 - T_2}{T_6 - T_2} \quad \text{(Figure 5)}$$

(i.e. turbine-inlet) temperature (T_4—Figure 5) will be lower than the combustion-chamber outlet temperature (T_6—Figure 5). This will result in a loss of thermal energy and a reduction in efficiency.

A considerable improvement in thermal efficiency can be obtained by splitting the heat exchanger of Figure 5 into two sections (Figure 6).

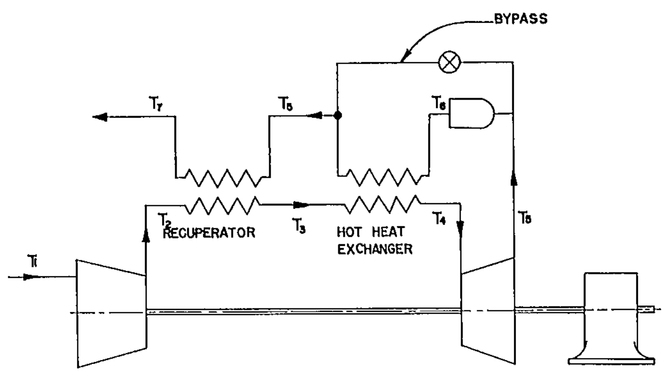

Figure 6. Improved exhaust-heated cycle.

The flow is arranged so that only a fraction of the turbine exhaust is passed through the combustion chamber and through the high-temperature section of the heat exchanger, where the gases are cooled to the turbine-exhaust temperature. At this point the gases are allowed to mix with the remainder of the turbine-exhaust air and together pass through the low-temperature part of the exchanger. In this particular case separate exchangers are used for the high- and low-temperature sections. The high-temperature exchanger is called the hot exchanger and the low-temperature exchanger the cold exchanger.

The thermal efficiency of the split cycle depends primarily on the thermal ratio of the 'cold' heat exchanger and the turbine-inlet temperature, and a plot showing the comparison between the simple exhaust-heated cycle and the improved cycle is given in Figure 7.[4]

It must be borne in mind that the efficiency curves plotted here do not take into consideration any combustion losses which were impossible to predict at that stage. The efficiency figures quoted must be multiplied by the combustion efficiency in order to obtain the actual thermal efficiency of the cycle.

All the aforementioned calculations showed that the exhaust-heated cycle could be quite attractive, provided that good combustion efficiencies were obtainable.

[4] A detailed analysis of the performance of the improved exhaust-heated cycle was made for a wide range of operating conditions in order to assess the effect of representative variables such as pressure losses, temperature ratios, thermal efficiency and specific power output. This work is reported on in References 4 and 5.

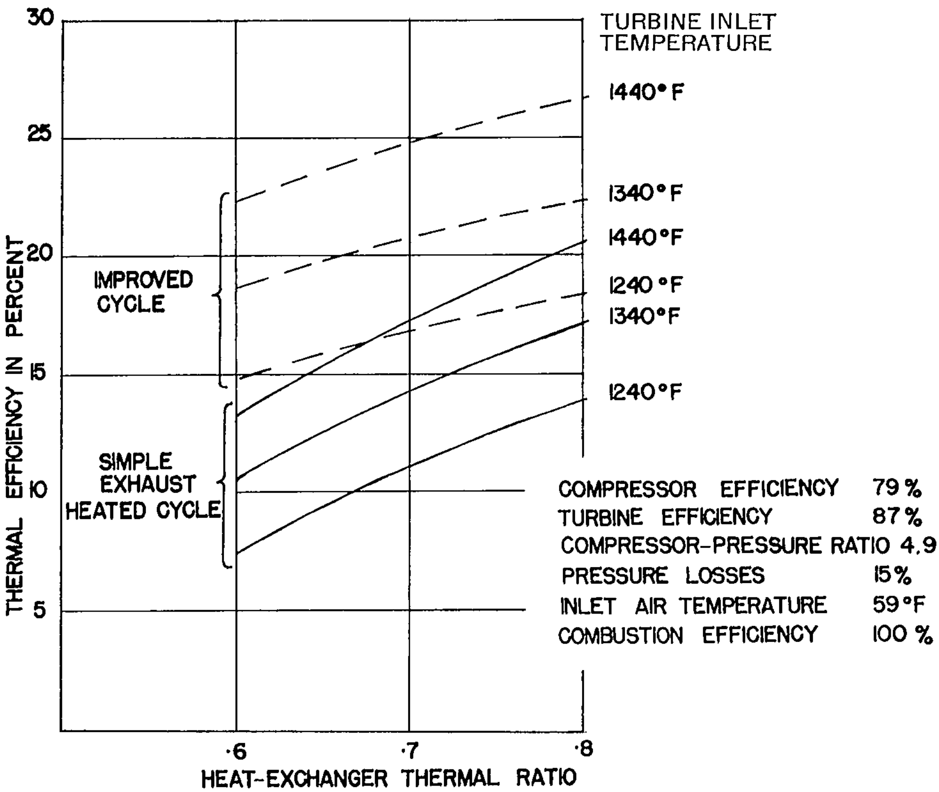

Figure 7. Cycle thermal efficiency vs. heat-exchanger thermal ratio.

The actual size of the equipment required would depend mainly on the power output and the thermal efficiency desired. The latter consideration, together with the pressure-loss allowance, would determine the surface area and hence the bulk of the heat exchangers.

PRELIMINARY PLANNING and DESIGN CONSIDERATIONS

• General Considerations

The use of an exhaust-heated cycle held out the promise of complete elimination of any turbine troubles (since the turbine would be using pure air), but its success would depend on the development of a high-temperature heat exchanger and an efficient combustion system.

Consequently, in planning the test facility, major stress was laid on the design of those two components, while standard and inexpensive equipment was used as far as possible in other parts of the installation.

Thus, when Rolls-Royce Ltd. (Derby, England) made available without cost an early version of the Dart turbo-propellor engine, it was decided to use this engine in spite of the fact that an aeronautical gas turbine was not the type best suited to this kind of work, owing to its lightness and relatively short life expectancy at design conditions.

In order to overcome this last limitation it was proposed to 'de-rate' the engine and run it at speeds and temperatures considerably below the design figures.[5] (This, needless to say, would reduce the over-all efficiency and power output considerably.)

Considerable savings were also effected in constructing the coal-handling system by using, as far as possible, equipment generously donated by J. I. Yellott, director of the research program conducted in the United States by the Locomotive Development Committee of B.C.R.

• Determination of Design Conditions

The choice of the Dart engine for the power unit immediately determined the cycle temperatures and pressures and provided the information required to proceed with the detailed design of the equipment.

This work commenced late in 1950. Performance curves of the Dart engine, supplied by Rolls-Royce, were used in the design, but every attempt was made to simulate, wherever possible, the conditions which would be encountered in a similar unit for a 4,000-hp engine, maximum usefulness from the experimental results being thus assured.

The improved cycle shown in Figure 6 was chosen as the basis for the thermodynamic design, and a preliminary performance analysis, made with the compressor and turbine characteristics of the Dart RD-A4 engine operating at 13,500 rpm, gave the following results (7):

Compressor-intake temperature	59°F
Compressor delivery pressure	64 psia
Compressor air flow	14.6 lb/sec
Turbine-inlet temperature	1,340°F
Turbine-inlet pressure	62.1 psia
Over-all heat-exchange thermal ratio	0.6
Pressure losses, air side (assumed)	2%
Pressure losses, gas side (assumed)	9%
Ideal thermal efficiency (100% combustion efficiency and no heat loss)	16%
Ideal power output	510 hp

[5] A turbine inlet temperature limit of 1,350°F and a maximum speed of about 13,500 rpm were decided upon as compared with normal maximum conditions of 1,000 hp output at 1,520°F and 14,500 rpm. The power output under these restricted conditions would be reduced to below 500 hp.

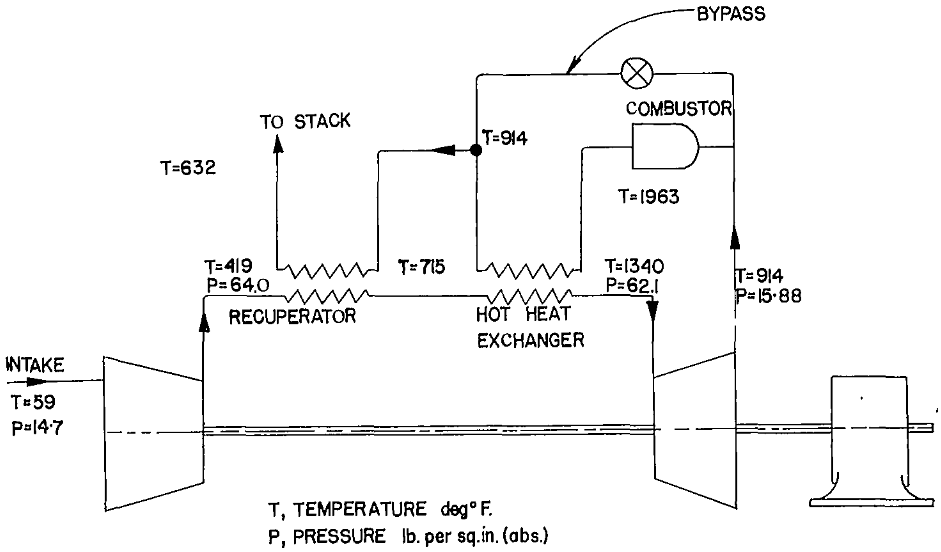

Figure 8. Thermodynamic-cycle diagram.

The temperatures and pressures obtained at various points of the cycle are shown in Figure 8.[6]

These figures were based on an over-all thermal ratio of 0.5 and total cycle-pressure losses of 11 per cent, the air-side loss being held down to 2 per cent (1.28 psi), while 9 per cent (1.45 psi) was allowed on the gas side.

An exhaustive study was made of the effects on size and efficiency of various thermal ratios and pressure losses in an industrial 4,000-hp locomotive unit (4) (5), and losses of about 12 per cent with a thermal ratio of 0.6 were found to result in the best compromise between cost and size on one hand and power and efficiency on the other.

Since the pressure ratio of the Dart-engine compressor would be lower than that of an optimum industrial unit (4.3 as against 5.9), a total loss of 11 per cent was finally allowed for the test installation.

Once the cycle conditions were decided on, detailed design of components was undertaken. The most difficult, and at the same time the most important, of these was unquestionably the hot heat exchanger. Here the designer found himself very often in completely virgin design territory, as there was no information in the literature about any exchangers in the world operating at such high metal temperatures.

Thus, problems in design which are normally ignored or circumvented by using 'conservative' procedures had to be thoroughly examined and investigated, and special design procedures and novel solutions had to be evolved and perfected. In view of the importance of this phase of work, the design procedures are described here in some detail.

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[6] For details of cycle calculations see Appendix 1.

DESIGN OF HOT HEAT EXCHANGER

• Design Specifications

	Hot Stream	Cold Stream
Fluid circulated	Combustion gases	Air
Fouling and corrosion properties	Fouling, corrosive	Non-fouling, non-corrosive
Flow, lb/sec	8.0	14.6
Temperature in, °F	1,965	715
Temperature out, °F	914	1,340
Operating pressure, psia	16	64
Heat duty 2,300 Btu/sec		

• Design 'Philosophy'

The hot heat exchanger would operate at temperatures which are considerably higher than in conventional industrial practice. It would use such products of combustion as the hot fluid, and the possibility of loss of performance by fouling had therefore to be seriously considered. Also, since high-sulphur coals were to be used, there was a strong possibility of corrosive attack on the tubes. Although the pressures were moderate, the high metal temperatures required careful consideration of the creep strength of the material, which would result in a drastic reduction of allowable stresses.

The first decision that had to be made concerned what kind of exchanger to use, and a shell-and-tube heater was chosen as the most economical type.

The next question was whether to route the combustion gases through the tubes or through the shell side.

All mechanical considerations pointed to the use of an air-tube heater. If this were chosen, the low-pressure fluid would pass through the shell side and a shell and nozzles of lighter gauge could be used. This would result in a considerable saving, although it would probably be necessary to provide means of insulating the shell from the hot gases. The fact that the shell need not be cylindrical might prove an advantage in some applications (in a locomotive, for example).

Also, the hot-end tube plate would not be exposed to the direct impingement of hot gases and might not require cooling.

Since the distribution of the fluid flowing inside the tubes (in this case cold air) is usually good, uniform heat transfer coefficients on the cold air side are obtained. A poor distribution of flow on the hot gas side caused, for instance, by stagnant-gas pockets behind baffles, would then result in local lowering of tube-wall temperatures. This, of course, would be highly desirable. (The reverse is unfortunately true in the case of the gas-tube heater.)

While these advantages were fully realized, the problem of fouling dictated the adoption of a gas-tube design.[7] Little was known at the time about fouling at such high temperatures, as no industrial equipment operated at comparable levels. It was feared that frequent cleanings would be necessary, and easy access to the tubes was thus one of the primary considerations. This was reflected in the design and final layout of equipment.

Another problem considered was that of corrosion. Since the sulphur content of some of the coals could be rather high, some corrosion troubles were expected. However, since sulphur attack becomes serious in reducing atmospheres only, it was thought that the large excess of air present in the exhaust gas stream would provide an oxidizing environment which would considerably reduce the rate of attack. In any event this was the type of problem that could be successfully investigated on a full-scale test rig only, and it was for the express purpose of obtaining solutions to such problems (should they arise) that the experimental work was undertaken.

• Thermal Design of Hot Exchanger[8]

1. Temperatures

In view of the large temperature ranges of both fluids, a counterflow[9] arrangement was mandatory. The size of the exchanger would then be determined primarily by the pressure losses that could be tolerated. Of the total cycle losses, 40 per cent were arbitrarily allocated to the hot heat exchanger.

Although a pure-counterflow arrangement was thermally most economical, an examination of Figure 9(a) will reveal that the tube-wall temperature (shown dotted), which depends on the individual heat-transfer coefficients inside and outside the tubes would reach 1,560°F at the hot end.

If, owing to the formation by the baffles of stagnant flow regions, the air-side coefficients were locally reduced to less than the design value, the tube-wall temperature curve would be shifted upwards and could approach dangerously close to the maximum gas temperature of 1,965°F. To reduce the maximum tube temperatures, the hot exchanger was split into two sections, parallel flow being adopted in the high-temperature section.

The temperatures resulting from this modification are shown in Figure 9(b). The maximum calculated tube temperature is now 1,471°F, and the danger of higher local temperatures is somewhat reduced.

[7] A similar decision was made by C. A. Parsons and Co. Ltd. in England when designing the heater for their exhaust-heated gas-turbine locomotive unit.

[8] For a detailed mathematical account of the design procedure reference should be made to Appendix 2 and References 7 and 8.

[9] In a counterflow exchanger the two fluids flow in a parallel stream outside and inside the tubes, but in opposite directions. Thus the hot-fluid inlet is placed at the same end as the cold-fluid outlet, and maximum heat recovery is possible. In parallel-flow exchangers both fluids flow in the same direction.

2. Transfer of air from counterflow to parallel-flow section

The actual tube-length at which the transfer from counter to parallel flow should be made was arrived at by balancing the heat-transfer-surface requirements in each section against the pressure drop and the forced-convection coefficients obtained with the given configuration.

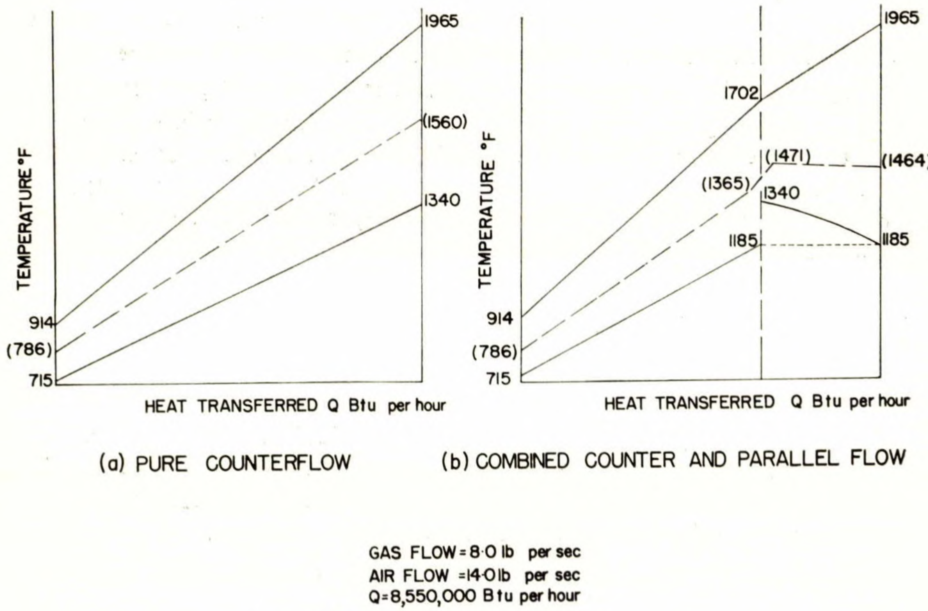

Figure 9. Hot-heat-exchanger temperatures.

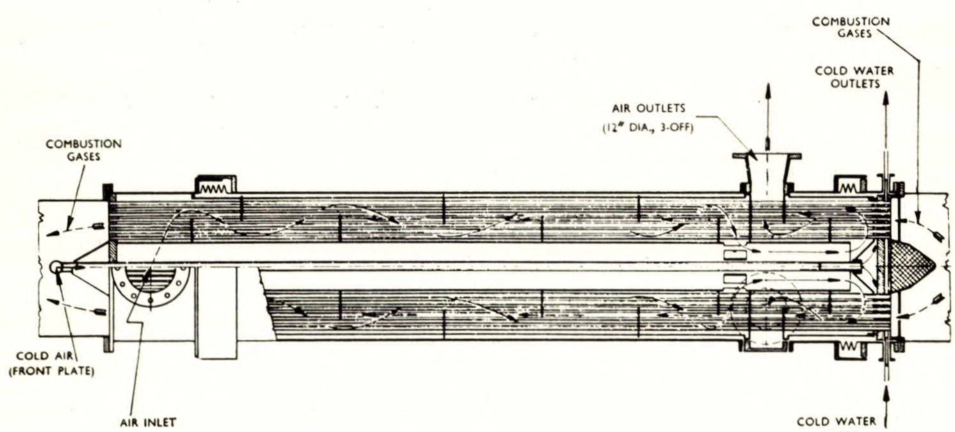

Figure 10. Arrangement of hot heat exchanger.

To save manufacturing expense, both sections were contained in a single shell and the air was transferred from the counterflow section to the parallel section by a central duct (Figure 10). The central duct had the disadvantage of making the shell diameter and the tube plates slightly larger, the result being a corresponding increase of the tube-plate stresses, but it facilitated a symmetrical flow pattern in the high-temperature parallel-flow section and accomplished the transfer with a minimum of loss in pressure and heat.

3. Calculation of heat-transfer coefficients and pressure drops

The coefficients on both sides of the exchanger were calculated by using the tube-flow correlations (with appropriate equivalent diameters) recommended in McAdams' *Heat Transmission* (9).

It was expected that the calculated value would be on the conservative side since no allowance was made for the increase in the coefficient which would probably result from the presence of tube supports in the shell.

The pressure drop on the shell side was also computed by means of longitudinal-flow correlations. This assumption, based on accepted procedures for exchangers with half-circle tube supports (10), simplified considerably the lengthy trial-and-error calculations required to design a suitable parallel-counterflow exchanger, but it resulted in an underestimation of the shell-side pressure drop as revealed by subsequent test results.

4. Size and surface requirements

As a result of several trial-and-error calculations matching the heat-transfer and pressure-drop requirements with various thermal-ratio and size estimates, the following figures were arrived at for the hot heat exchanger:

Shell size	38 in. I.D.
Tubes: number	498
size	1 in. O.D.
wall thickness	0.035 in. (20 gauge)
length	18.1 ft
Total heat-transfer (based on tube O.D.)	2,387 sq ft
Parallel-flow-section length	3 ft
Counterflow-section length	15.1 ft
Over-all thermal ratio	0.47
Air temperature at transfer point	1,185°F

• Mechanical Design

1. General considerations

The high operating temperature encountered in the exchanger made the problem rather specialized, and therefore most of the mechanical design was done in the Gas Dynamics Laboratory, where experimental work could be carried out on a small scale to test unusual design features.

The completion of the thermal design was followed by the preparation of a 'preliminary layout' drawing, which was submitted to several leading manufacturers for a quotation as to the approximate cost of fabricating and delivery time that could be expected. The design was then discussed with representatives of the firm which was given the contract[10] to complete the design and fabricate the heat exchanger.

It was mutually agreed that the responsibility for the thermodynamic design and proportioning of major components would rest with the Gas Dynamics Laboratory, while the manufacturer would be responsible for the design of such details as expansion allowances and assembly considerations. As the fields of design overlapped, close liaison was maintained for the duration of the work.

2. High-temperature tube-plate design

The high-temperature tube plate was the most critical component in the exchanger owing to the exacting design requirements imposed on it by the severe operating conditions to which it would be subjected. It is in close proximity to the high-temperature (1,965°F) gases leaving the combustion chamber and is subjected to a pressure of nearly 50 psi by the fluid surrounding the tubes. For these reasons, the material of the tube plate must have a high creep and rupture strength at elevated temperatures. One of the materials which satisfies these requirements and which was readily available is Type 310 (25% Cr—20% Ni) stainless steel, which was used in this instance. Even then the stresses tended to be excessive unless the temperatures could be reduced considerably.

This was accomplished by using a double tube-sheet with provision for passage of a coolant (air or, if required, water) between the two plates and along the rear surface of the main tube plate, Figure 11(a). The theory was that the thin auxiliary front plate (exposed to the high-temperature gas on one side) would not be required to carry any pressure stress at all (the pressure of the coolant in the central passage being approximately the same as that of the gas) while the main tube-sheet, which would carry the full pressure load, would not be exposed to the highest cycle temperatures.

The temperature levels, which would still be very high if air were used as a coolant, had to be known accurately because of the high sensitivity of creep and rupture strength to temperature.

Thus the problem of designing the main tube plate was divided into two parts: stress analysis and temperature analysis.

(*a*) *Stress analysis*—The pressure-stress distribution in the tube plate was calculated by using as a basis formulae for simply supported circular plates and modifying them to take into account the stiffening effects of the tubes and the effects of the tube holes in the plate (7).[11]

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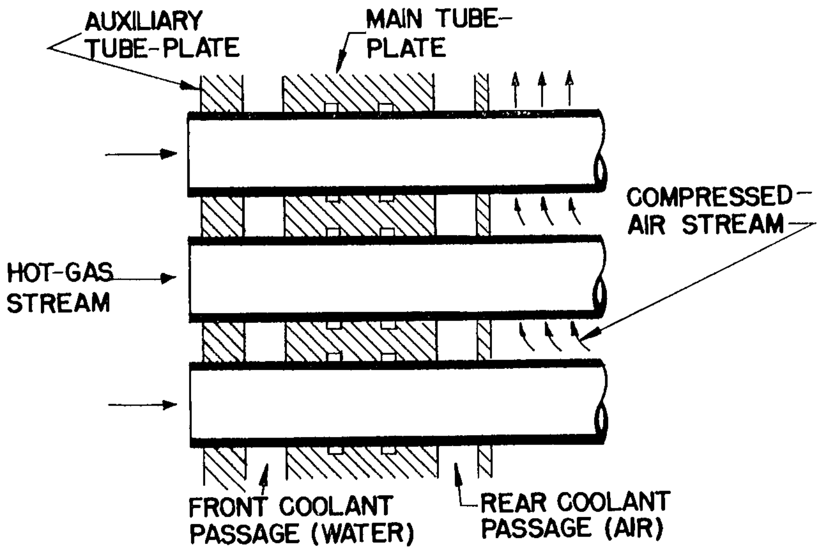
[10] Canadian Vickers (Canada) Ltd. were awarded the contract of fabricating both the hot and the cold exchanger and the turbine inlet manifold.

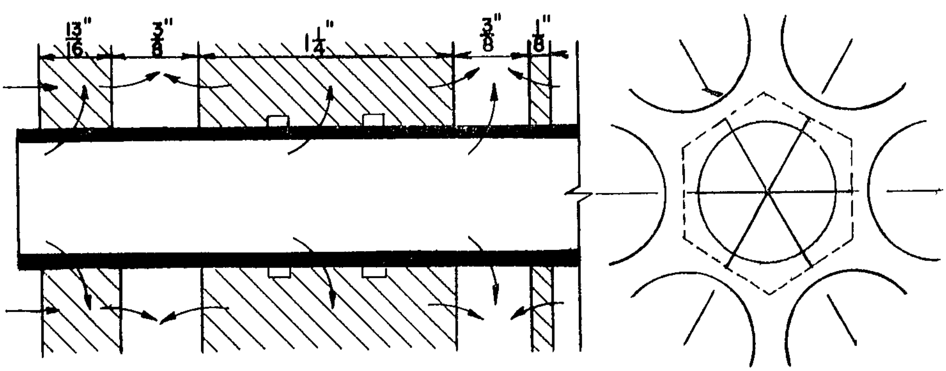
[11] For a more detailed treatment of the stress analysis see Appendix 2.

The solution of the temperature field enabled the thermal stresses to be computed according to the method of Timoshenko (11), and the two stress patterns were superimposed to obtain the combined radial and circumferential stress distributions.

The equivalent maximum stress was calculated by the maximum-shear-strain energy theory as recommended for combined stresses under creep conditions (12). With a cooling air flow of 0.05 lb/sec in each coolant passage, the maximum



a) TUBE-PLATE CONSTRUCTION



b) HEAT-FLOW PATH WITHIN TUBE SHEET

Figure 11. Hot tube-plate arrangement.

equivalent tube-plate stress was calculated to be 7,840 psi with a thickness of 1.75 inches and a maximum temperature of 1,425°F. Further calculations showed that use of water in the front cooling passage would result in the reduction of the maximum tube-plate temperature to less than half this value. This method of cooling was therefore adopted.

The auxiliary tube-sheet was made of Type 442 stainless steel. A major factor in the choice of ferritic Type 442 steel for this application was the need to minimize the differential expansion between the auxiliary and the main tube-sheets. The coefficient of expansion of ferritic steel is about one half that of austenitic steel, such as Type 310. Thus, temperature levels prevailing in the auxiliary sheet, while considerably higher than those in the main sheet, would not cause excessive differences in expansion between the two.

(*b*) *Temperature analysis*—The analysis of the temperature distribution within the tube-sheet was complicated by the three-dimensional nature of the heat flow. Heat enters the auxiliary tube-sheet at the front face and throughout the thickness of the plate at each tube hole. It is dissipated to the coolant through the rear surface, Figure 11(b) (6).

In the main tube-sheet heat is gained through the tube-hole surface only and dissipated at the two surfaces exposed to the coolant. If the temperature of the gas entering adjoining tubes is the same and the temperature of the coolant in the passage is uniform, there will be no heat transfer across the boundary between the hexagonal metal elements associated with each tube, Figure 11(b) and the temperature distribution will be exactly the same in each such element. This assumption, closely borne out in practice, formed the basis for establishing the heat-balance equation in differential form (Appendix 2). The equation was rather awkward to integrate and was simplified by assuming that there was no radial heat flow within the metal surrounding each tube but allowing for radial heat flow into the metal. This introduces a slight inaccuracy, which can be corrected approximately once the axial temperature distribution is obtained.

The solution of the differential equation required a knowledge of the convective heat-transfer coefficients of the gas and coolants.

The coolant coefficients were calculated by the use of accepted correlations for flow in narrow passages.

The usual method of calculating the gas coefficient could not be used, as conditions of fully developed tube flow do not exist in the short entrance length under consideration.

Although it was well known that the coefficients are much higher in the inlet region (theoretically tending to infinity at the sharp-edge inlet to the tubes, where there is practically no boundary layer to resist the heat flow), the only correlation available was confirmed experimentally at low temperatures only.

Since the accurate determination of the entrance-length gas coefficient at high temperatures was of paramount importance, a special small-scale test rig was designed and constructed in order to investigate this problem.

One of the most successful outcomes of this investigation was the development of a liner which could be inserted into the tube entry to reduce the heat flow into the tube plate and to protect its front surface from ash-laden combustion gases. Although stainless-steel inserts with a stagnant-air gap between the liner and the tube, Figure 12(a) proved most effective in reducing the heat transfer, [12] ceramic liners, Figure 12(b) proved to be the most economical to fabricate and were specified in the final design.

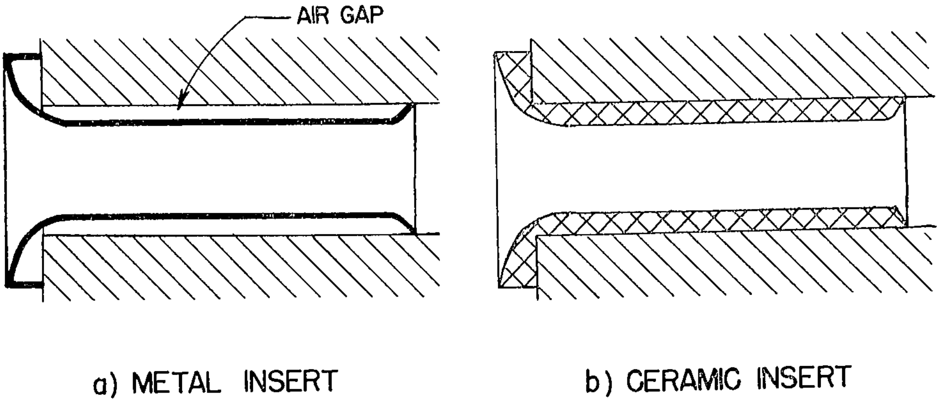


Figure 12. Tube liners.

Once the figures for entrance-length coefficients were available, the complete solution of the temperature field was obtained. It is shown graphically in Figure 13(a), for air-, and in 13(b) for water-cooled tube-sheets. The dotted line in Figure 13(a) indicates the temperature with the correction for radial heat flow included.

The maximum temperature and the surface temperatures thus obtained were used in computing thermal stresses. Linear temperature gradients were assumed between the surface and the point of maximum temperature (as shown by the broken lines in Figure 13(a), as otherwise the equation could not be readily integrated. Figure 13(a) shows that this is a tenable assumption. On the basis of metal temperatures thus found, it was decided to employ water as the cooling medium in the front passage.

3. Tube bundle design

The components subjected to the highest metal temperatures in the exchanger are the tubes. Calculations showed that within the first three feet of length (i.e. in the parallel-flow section) the tube-wall temperature would be fairly uniform at about 1,470°F.[13]

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[12] These conclusions, originally obtained in limited tests at the Gas Dynamics Laboratory, were fully confirmed by more extensive experiments carried out at the Fuel Research Laboratory of the Department of Mines and Technical Surveys. The Department tests also showed that there appeared to be no decrease of heat-transfer coefficient due to fouling and that the ash deposit could be removed readily from the tube surface.

[13] See page 6 of Reference 7.

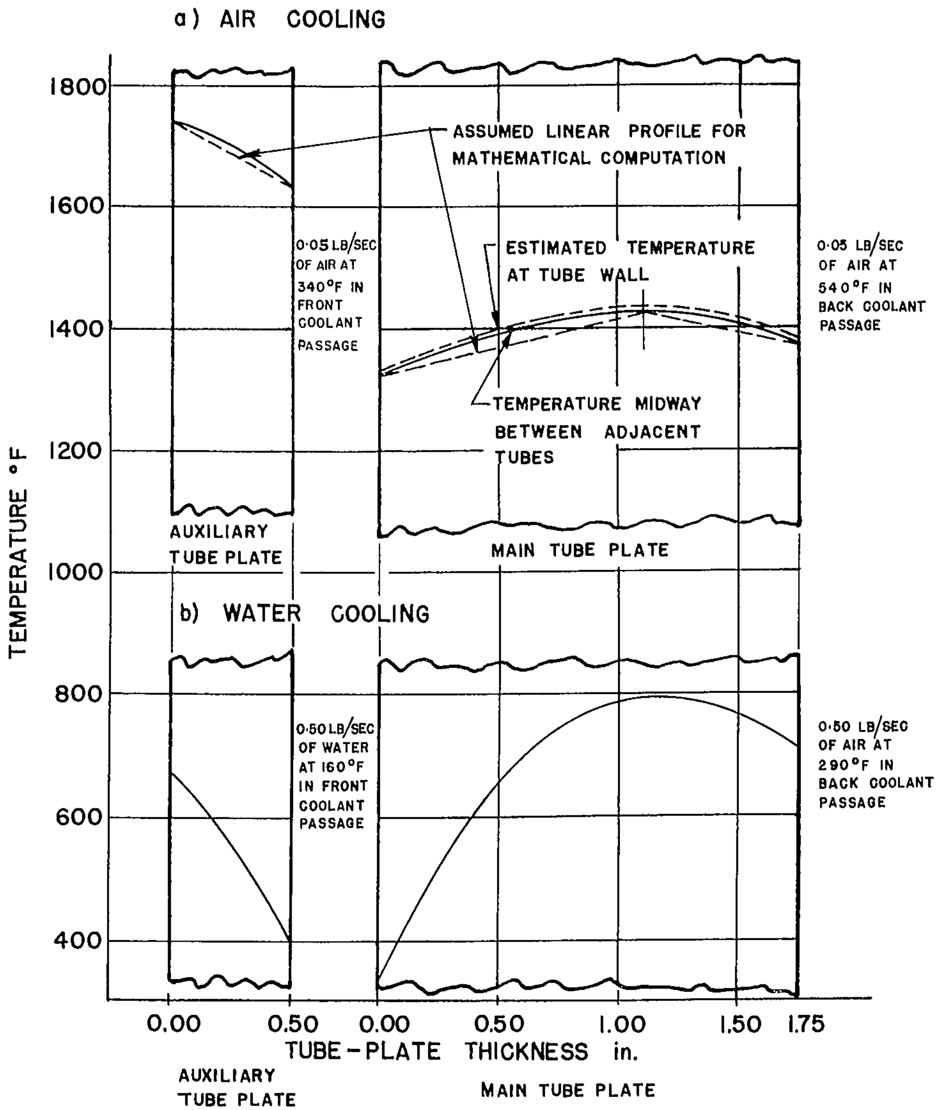

Figure 13. Temperature distributions in tube plates. (Exhaust-gas temperature = 1965°F)

Stainless steel would be required at those temperatures and AISI Type 347 or 321 seamless tubing was specified originally. However, owing to a serious shortage of high-grade steel which prevailed in Canada and the United States at that time, it was impossible to obtain delivery dates that would not delay the whole program considerably. Thus, when it was found that Nimonic 75 tubes were readily available in the United Kingdom, the decision was made to accept them.

Although it was known that they might prove more susceptible to sulphur attack than the types specified, refusal to accept them as substitutes would have delayed the experimental work by at least a year.

Tube stresses were calculated by two different methods (7) and were found to be such as to produce a creep rate of about 0.002 inch on the diameter per 1,000 hours of service, with a safety factor of about 4.7 on the rupture strength. Although this would not be acceptable in commercial service, it was considered adequate for experimental purposes. A reduction of tube stress could be readily effected by increasing the tube-wall thickness, but this would have entailed a considerable increase in the cost of the tube bundle.

4. Air-outlet nozzles

The design of the air-outlet nozzles was governed by considerations of pressure loss, tube temperature and space limitation.

To keep the discharge velocities down and at the same time to insure a good flow distribution and effective use of the parallel-flow section, three outlet nozzles were provided (at intervals of 120° around the circumference). The top nozzle fed three turbine inlet ports through a short 12-inch-diameter duct, and the two remaining nozzles fed two inlet ports each through somewhat longer 12-inch ducts.

5. Shell design

The shell was provided with an inner liner and air cooling in order to reduce the shell temperature sufficiently to permit the use of carbon steel. The cooling air, bled from the Dart compressor delivery and admitted at the intermediate tube plate, reduced the maximum temperature of the shell to 700°F. Since both tube-sheets were fastened rigidly to the shell, two bellows-type expansion joints were used. The shell was suspended from the mezzanine steelwork so as to allow some freedom of movement and thus reduce stresses set up by the expansion of various parts of the equipment when it is operating.

FURNACE DESIGN

• General Considerations

The component of the plant which underwent the greatest evolution and was subjected to the largest number of modifications was unquestionably the furnace. In all, three different designs were tested, the last one being radically different from the original.

Since all of the modifications made at various times resulted from actual operating experience, they will be described in detail in subsequent chapters dealing with the experimental work.

At this stage general design considerations are given, together with the description of the unit used in the first series of tests of the complete plant.

• Design Requirements

The primary requirements of the furnace were as follows:

(*a*) It should burn coal, crushed to ⅛-inch mesh size, the combustion taking place at a pressure slightly above atmospheric with about 100 per cent excess air.

(*b*) Highly preheated air (up to 900°F) was to be used for combustion.

(*c*) The pressure drop across the furnace should be a minimum.

(*d*) Combustion should be efficient, with minimum carry-over of ash. Slag should be removed automatically.

Owing to the high preheat temperatures, it was deemed advisable not to use movable parts within the furnace.

Tests conducted at that time in the Fuels Research Laboratory, Mines Branch, Department of Mines and Technical Surveys, on a ~~6~~ 24-inch slagging cyclone combustion chamber suggested the feasibility of using this type of furnace for the exhaust-heated cycle.[14]

Its simplicity, the absence of moving parts and its capability of disposing of the slag continuously, made it an attractive possibility.

Considerable operating experience with slagging cyclone furnaces was accumulated by the Babcock and Wilcox Company, but it related mostly to much larger sizes (of the order of 4 to 8 feet in diameter) and their work indicated that small units, designed simply by scaling down, did not operate satisfactorily.

Thus an appreciable amount of development work was anticipated, and the design and construction of the furnace was given high priority with the purpose of having the unit complete at an early date. In this way, considerable running time and experience would be accumulated before the heat exchangers were ready for installation.

• Preliminary Work

Consideration of heat-release rates showed that a 24-inch-diameter furnace would be required for the exhaust-heated turbine, and the first unit was built and tested by the Department in Ottawa. When this work was interrupted for structural reasons (not connected with the test program), another unit was designed and built in the Gas Dynamics Laboratory on the basis of the results obtained in the preliminary tests in Ottawa and information supplied by Babcock and Wilcox.

[14]The principle of operation of this furnace is as follows: Air enters the furnace tangentially, creating a strong vortex flow inside the cylindrical section. Coal is injected into this 'cyclone' and thrown on the hot walls by centrifugal force. As the coal burns, the ash residue forms a layer of molten slag along the walls. Coal particles embed themselves in this layer and are burnt by the air flowing past the surface. The molten slag flows slowly towards an opening in the wall through which it is continuously drained. Some fines in the coal burn in the air stream without reaching the wall and the resulting fly ash passes out with the exhaust gases, but when the furnace operates properly, a very large percentage of the ash is removed as slag.

This furnace was first run on December 10, 1952, and about 40 hours of operation were achieved[15]—about 25 hours on coal—when a minor failure in the Dart turbine, which was being used as a slave, interrupted the program.

By that time, however, enough information had been obtained to proceed with the design of a unit for the complete plant.

• Description of Furnace

The furnace consisted of two sections: a water-cooled 24-inch-diameter primary conbustion zone and a mixing section (Figure 14).

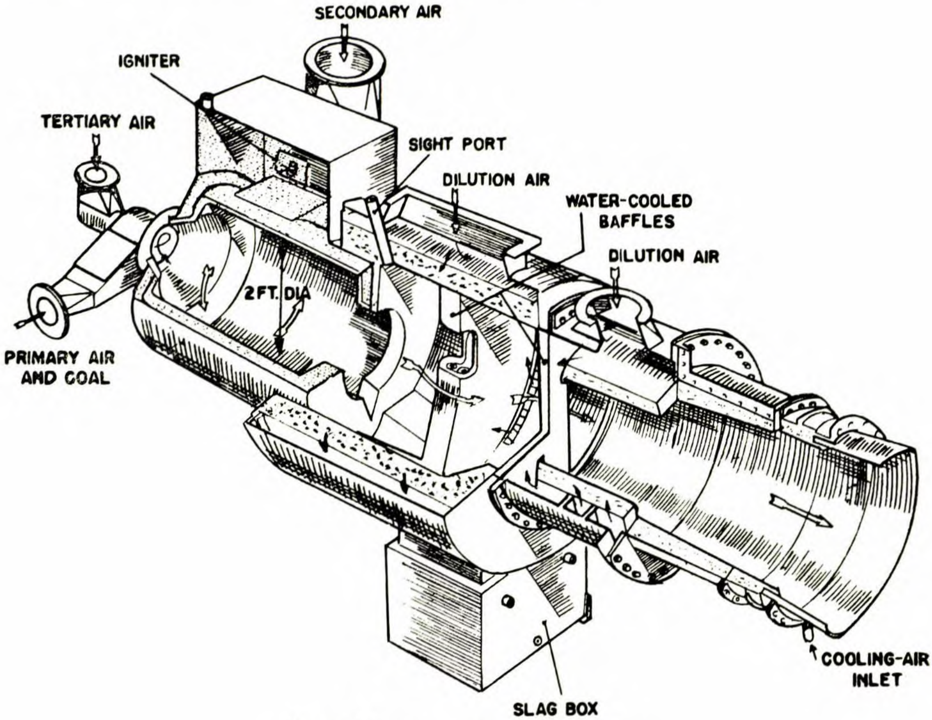

Figure 14. Furnace (first series of tests).

1. Primary combustion zone

The primary zone was a slightly modified Babcock and Wilcox design, sloped at 5 degrees to the horizontal and separated from the mixing zone by an orifice-type throat with a cut-out to facilitate slag-draining. The Babcock-Wilcox designs featured a re-entrant throat at this section, but as this would induce an additional pressure loss, it was hoped that satisfactory combustion could be obtained without it.

Two air inlets were provided in the primary combustion (cyclone) zone apart from the primary air, which entered with the coal.

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[15] A more detailed description of these tests is given in Appendix 3 and in References 13 and 14.

They were arranged to discharge tangentially so that the swirling motion of the air would throw the coal particles to the wall by centrifugal force, where they would be deposited and burnt rapidly by the whirling air stream.

The slag which formed on the wall would move slowly downstream through the V cut-out in the throat and down to the slag hole.

The slag hole was placed downstream from the throat. The slag draining through it dropped into a slag pit situated directly underneath.

The pit was operated half full of water, with a continuous supply and an overflow system. The slag, which granulated into small pellets upon being quenched in the water, was periodically flushed out by water jets and collected on screen trays for weighing and analysis.

2. Mixing section

The primary purpose of the mixing section was to reduce the temperature of the gas by diluting it with a portion of the turbine-exhaust air. Two water-cooled vertical baffles were arranged in the mixing section to insure good mixing and eliminate direct radiation from the flame to the hot-heat-exchanger tube plate, which was directly downstream from the furnace.

Dilution air was introduced into the mixing section at two points. It was ducted in such a way as to provide some cooling between the inner brick wall and the outer steel shell.

The whole furnace assembly was mounted on sliding runners to permit freedom of expansion and easy access to the inside for maintenance and repairs.

3. Wall construction

The entire combustion zone was internally lined with plastic chrome-ore, a castable refractory material, which was mixed with water and rammed into position during construction. Three-quarter-inch-long steel studs were welded to the inside of the wall to help retain the refractory in place. The mixing section was lined with medium-quality fire-brick backed with insulating brick.

ARRANGEMENT OF EQUIPMENT

• Test Cell

Although the most promising application for the coal-burning turbine seemed to be a locomotive, no effort was made to simulate a locomotive layout in the actual arrangement of the equipment.

On the contrary, ample space was allowed everywhere so that each component would be easily accessible and changes could readily be made if necessary.

It was decided to locate the plant in the Gas Dynamics Laboratory at Macdonald College, Ste. Anne de Bellevue, where a space of approximately 2,500 square feet with 20 feet of headroom was available and all service facilities could readily be installed.

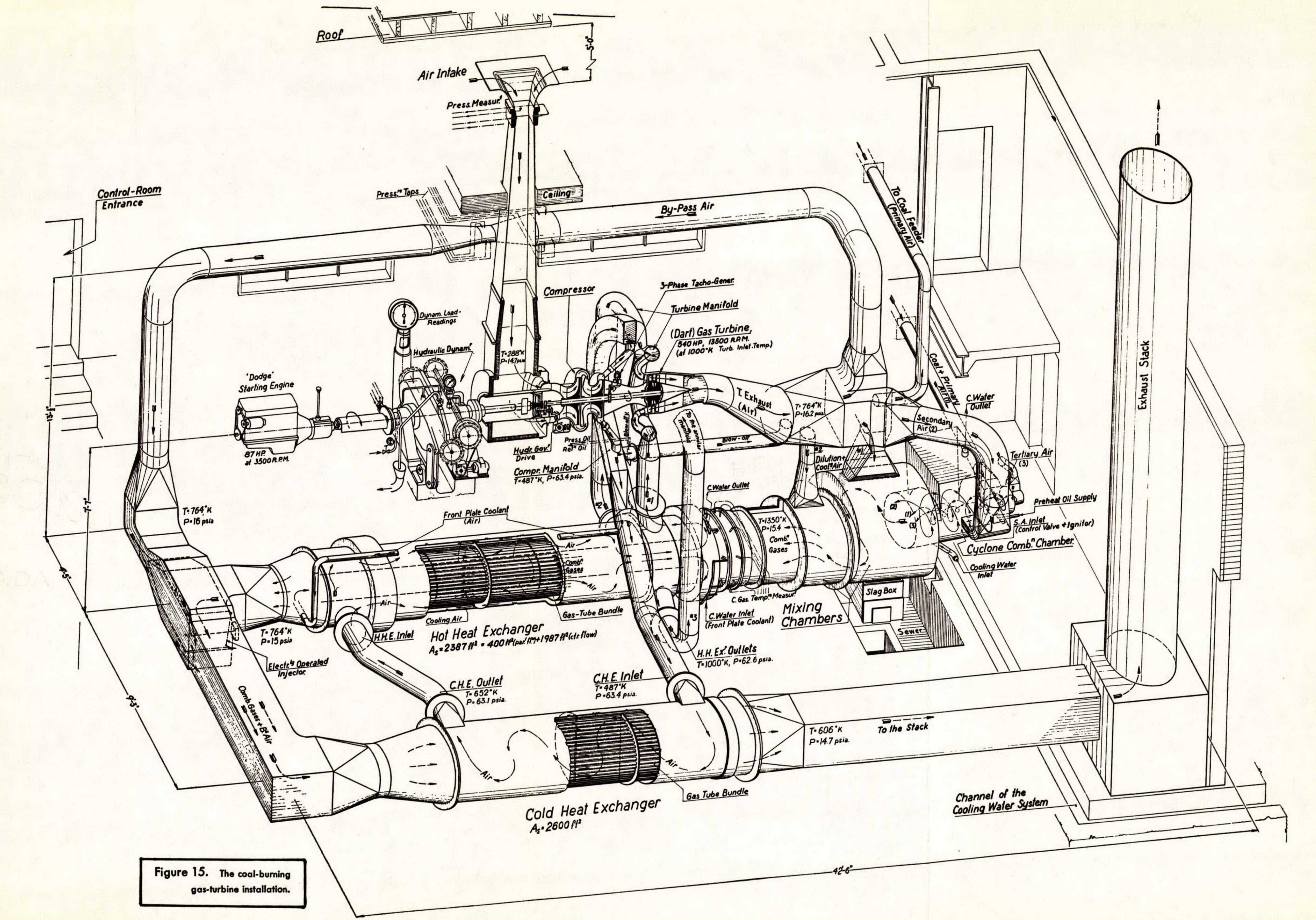

Figure 15. The coal-burning gas-turbine installation.

The test cell had a floor area of 30 by 50 feet and was enclosed by cement-block walls sound-proofed on the inside.

The control room was located alongside the test cell on the mezzanine floor level, with the coal-storage and coal-handling equipment in a fire-proof cell underneath.

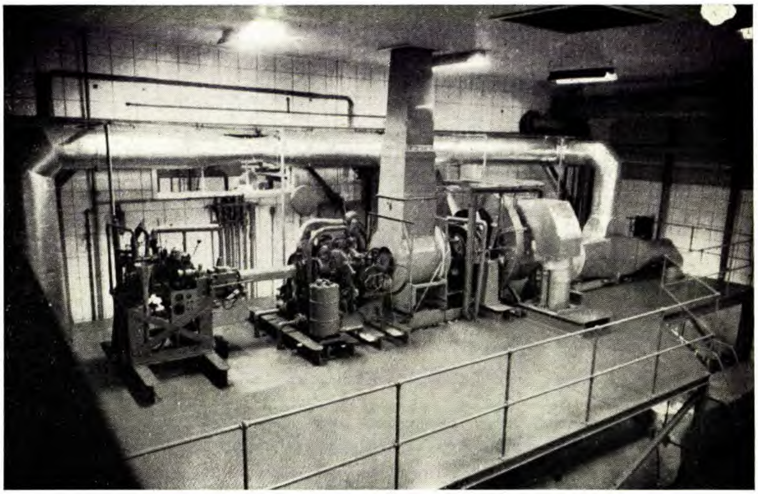

Figure 16.
Machinery line.

The actual layout of the equipment is shown in Figure 15. The starting engine-dynamometer-gas turbine assembly was located on a steel mezzanine floor (Figure 16), while the furnace and the heat exchangers were placed underneath (Figures 17 and 18).

As shown in Figure 15, the air was admitted from outside the test room through a calibrated 10.5-inch-square bell-mouth orifice containing four pressure taps for flow measurement. The air intake was surrounded by a system of baffles to reduce the noise level.

The air then passed via a plenum chamber into the Dart compressor,[16] where it was compressed to 64 psia. A special manifold was designed to collect the compressed air from the seven individual compressor outlets and duct it to the first, or 'cold', heat exchanger.

This exchanger was of conventional shell-and-tube, packed floating-head design, containing 760 one-inch tubes on 1¼-inch triangular pitch. It was made entirely of carbon steel, and the total heat-transfer surface was 2,600 square feet.

From the cold exchanger the air, now at 715°F, was admitted to the high-temperature ('hot') exchanger, where the temperature was raised to 1,340°F by the furnace exhaust gases flowing through the tubes. The air, emerging from the three nozzles provided at the outlet from the exchanger, was conveyed to the turbine through three vertical ducts and a specially designed inlet manifold.

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[16] Details of the engine are given in Appendix 4.

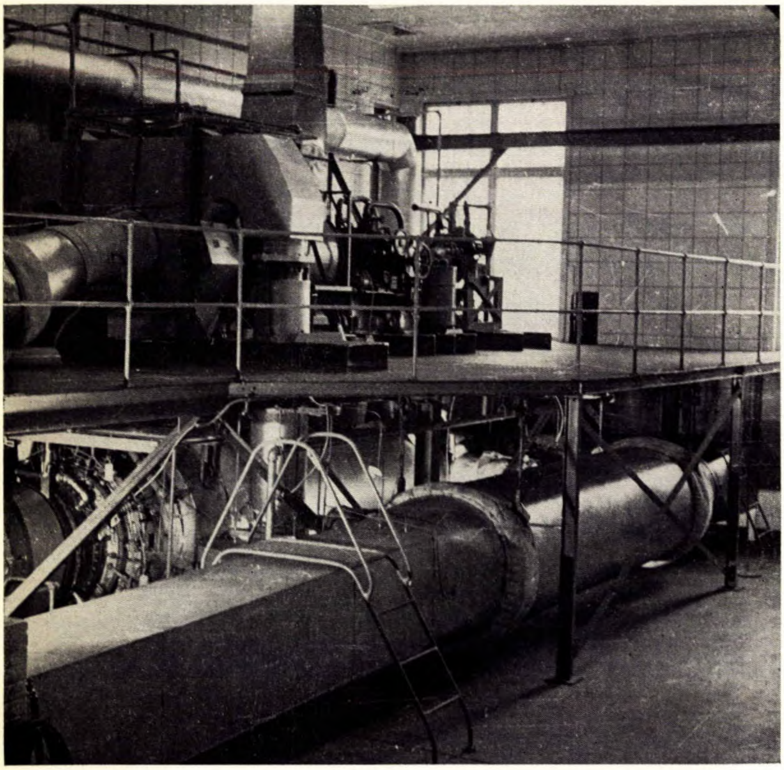

Figure 17.
Coal-burning turbine test cell (showing heat exchangers under the mezzanine floor).

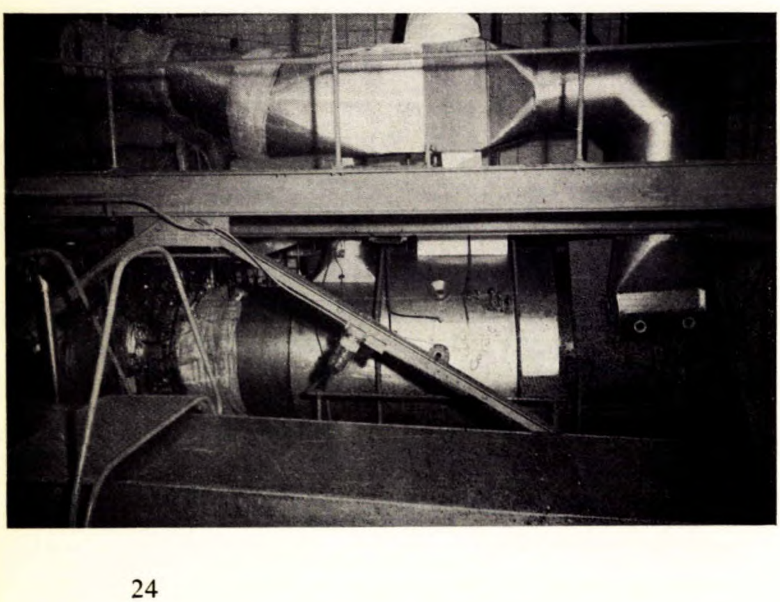

Figure 18.
Turbine-exhaust ducting and furnace.

The expanded air from the turbine was divided into two main streams:

(*a*) a combustion-air stream

(*b*) a bypass stream.

Stream (*a*) consisted of a number of individual streams that were admitted into the cyclone combustion chamber at various sections as dictated by the combustion requirements[17] and then allowed to pass through the tubes of the hot exchanger. Stream (*b*) was made to bypass the furnace and the hot exchanger. A venturi meter was placed in the bypass duct to measure the flow.

The two streams were brought together downstream from the hot exchanger, the slightly higher pressure of the bypass stream being used to produce an ejector effect which would offset slightly the furnace and hot-exchanger pressure losses. The ejector was fitted with an electrically operated variable throat so that the amount of bypass air could be regulated.

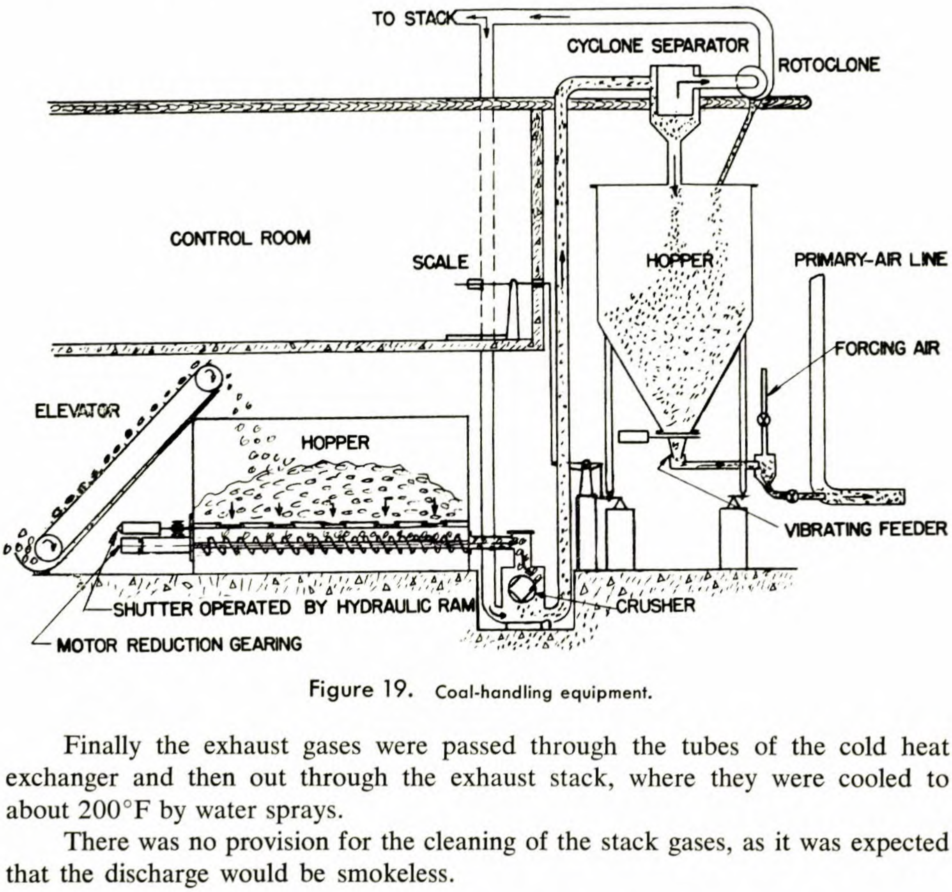

Figure 19. Coal-handling equipment.

Finally the exhaust gases were passed through the tubes of the cold heat exchanger and then out through the exhaust stack, where they were cooled to about 200°F by water sprays.

There was no provision for the cleaning of the stack gases, as it was expected that the discharge would be smokeless.

[17] See page 21 and Figure 14 for a detailed description of the furnace and location of air ports.

• Coal-handling System

The original coal-handling equipment is shown in Figure 19.

Coal from the coal-bunker was transported by an elevator to a storage hopper from which a feed-screw delivered it to a crusher.

Here the coal was crushed to a ⅛-inch mesh size and lifted by air suction to a cyclone separator, from which it fell by gravity into the weighing hopper. The suction was provided by a Rotoclone blower.

The second hopper was placed on a scale which was so arranged that the fuel weight could be read directly on a balance arm inside the control room.

A strain-gauge transducer was later developed which made it possible to obtain a permanent record of the coal weight consumed during a test.

The bottom of the hopper was closed by a ram-operated slide-valve which, when open, allowed the coal to drop into a tubular vibrating feeder. The vibrating feeder conveyed the coal into an injector, where it was picked up by a stream of pressurized air[18] and delivered to the furnace.

The metering was originally performed by a rheostat control on the vibrating feeder, but in later tests good regulation was obtained by varying the forcing air pressure.

INSTRUMENTATION

In order to obtain as complete information as possible about the performance of every component of the test plant, a major part of the original planning and design effort was devoted to the problem of instrumentation. As a result the plant was originally equipped with more than 190 thermocouples and pressure taps, and many more were added as the work progressed.

Table I gives a summary of the allocation of various instruments and controls.[19]

Table I

Summary of the Coal-burning-turbine Instrumentation

Location	Temperature Measurement	Pressure Measurement	Miscellaneous Instruments and Controls
Various cycle positions	51	26	
Hot heat exchanger	54	5	
Furnace	25	15	
Dart engine	4	5	5
Starting engine	4	2	6
Total	138	53	11

[18] In later tests the vibrating feeder was dispensed with and the coal was conveyed by means of compressed air directly from the hopper.

[19] A more detailed list of instruments is given in Appendix 5.

Such a vast array of instruments created the problem of grouping and arrangement to obtain the maximum of useful information during the operation of the plant. Consequently, the following plan was adopted:

(*a*) Instruments indicating readings which were critical in the operation of the plant were grouped around the 'driver's' seat at the control desk, which contained all the controls necessary for running the plant. The operator thus had an immediate indication of any malfunction of the equipment.

(*b*) Most of the instruments classified in (*a*) were also connected to precision electronic recorders so that a permanent record of the behavior of the plant would be available.

(*c*) The remaining instruments were read by the operating crew at 30-minute intervals throughout the test run.

• Temperature-indicating Instruments

Chromel-alumel thermocouples were used almost exclusively for temperature indication. They were shielded for radiation in high-temperature zones.

A platinum—platinum-rhodium thermocouple was used as a calibrating standard.

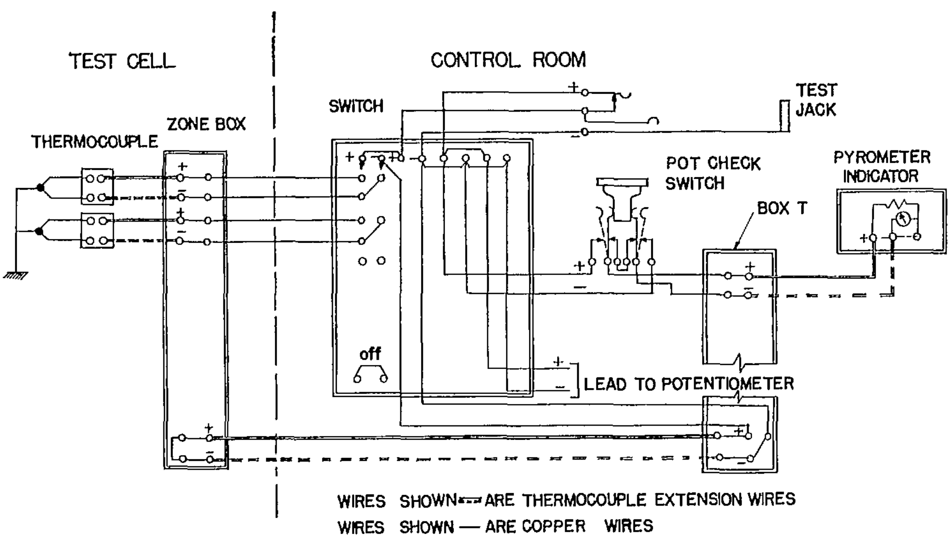

Figure 20. Thermocouple circuit.

In order to save expense, thermocouple leads were not run the full distance of 30 to 60 feet to the instruments but were grouped in zone boxes dispersed throughout the plant (Figure 20). Copper leads were used between the zone box and the indicator, together with a compensating thermocouple to offset the temperature difference between the zone box and the instrument.

To obtain maximum flexibility, the leads from the zone boxes were not connected directly to the indicators but were grouped in transfer boxes inside the control room. The leads from the indicators were connected to similar transfer boxes placed alongside. Connections then established by jumpers from one box to another made it possible to connect any thermocouple to any desired indicator without disturbing the instrument.

Three types of indicators were used:

(*a*) galvanometer-type indicators with 24-point switches

(*b*) potentiometer-type Brown electronic self-balancing precision indicators with 24-point switches

(*c*) potentiometer-type electronic precision recorders with 16-point circuit.

Whenever accurate knowledge of temperatures was required for performance calculations, the thermocouples were connected to the potentiometer-type precision indicators. Other temperatures were read on the galvanometers. Inaccuracies in reading the galvanometers introduced by varying circuit resistances could be taken into account by incorporating a special switch (Figure 20) which enabled the operator to switch any thermocouple from the galvanometer to the potentiometer circuit and thus obtain the galvanometer error by direct comparison of the readings.

Some of the temperatures (such as cooling-water, engine-oil, etc.), of which very accurate indication was not required but which nevertheless were vital to the proper operation of the plant, were obtained by means of electric resistance thermometers and read on aircraft-type indicators.

• Pressure-indicating Instruments

Most of the cycle pressures were read on water manometers. These were referred to atmosphere or, in the case of the high-pressure readings, to the Dart-compressor delivery pressure, which was read on a precision Bourdon gauge. Thus, small pressure losses of a few inches of water could accurately be measured at pressures of several atmospheres.

A special design was evolved which allowed quick assembly of large numbers of high-pressure manometers.

• Controls

The main controls consisted of the coal-feed control,[20] kerosene-flow control (when coal was not being burnt) and dynamometer-load control.

All these, plus the starting-engine controls and the various valves and switches for regulating cooling-water flows, were grouped at the driving desk (Figure 21) and were arranged so that the whole plant was always under the complete control

[20] Depending on the feed method used in different tests, the control was obtained by:
(a) a rheostat control on the vibrating feeder
(b) control of the forcing air pressure
(c) feed-screw speed control

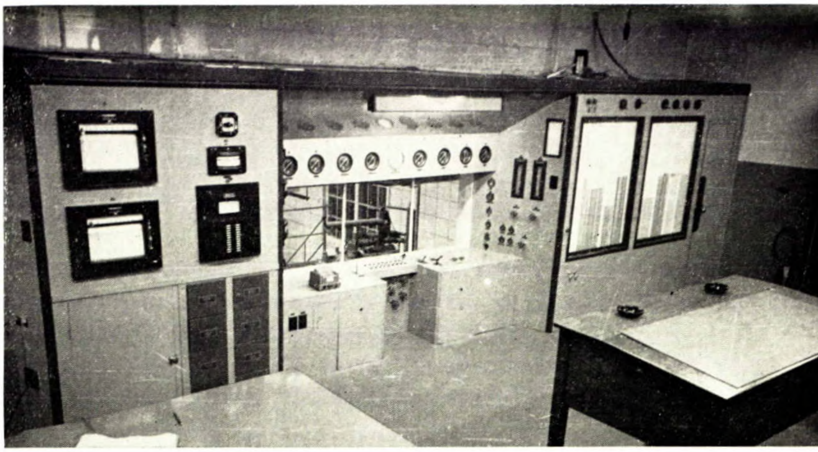

Figure 21.
Control room.

of one operator. An intercommunication system was developed which allowed direct communication with observers inside the test cell in spite of the high noise level prevailing there during the operation of the plant.

Since the main purpose of this work was to obtain information about the heat exchanger and the furnace, no effort was made to develop any automatic controls.

Overspeed protection was afforded, however, by an automatically operated valve arranged to spill compressor-delivery air directly into the furnace inlet, bypassing the turbine and thus protecting the engine in case of overfueling or failure of the dynamometer or water supply. The speed at which the governor would operate could be selected manually and an override was provided which would fully open the compressor blow-off valve and keep it open, thus stopping the plant in a few seconds in case of emergency.

During normal operation, small speed fluctuations were corrected by adjusting the dynamometer load, while major speed and temperature changes were obtained by adjusting the fuel flow.

PART **2**

FIRST SERIES OF TESTS (NOVEMBER 1953-APRIL 1955)

PRELIMINARY TESTS

The design, construction and assembly work reported in Part I continued throughout 1951 and 1952. The fabrication work was considerably delayed by steel shortage and by a prolonged (July to October, 1952) strike at Canadian Vickers Ltd., who were manufacturing the heat exchangers.

In order to accelerate the work, various components of the plant were rig-tested as soon as they became available. Thus when the Dart engine arrived from England in September 1951, it was installed immediately for preliminary proving tests even though the test-cell construction was not complete at that stage.

The original run, made in October, was limited to a few minutes, since the turbine exhausted directly into the room. By February 1952 construction of the test cell, including sound-proofing of the complete cell and of the air-intake system, was complete. Extensive tests were then run on the engine in its 'as-received' condition in order to obtain a complete calibration of the turbine set and instruments. The results of these calibration tests were used in predicting the performance of the complete plant (15) and are reported in Appendix 6.

The design, fabrication and assembly of various components of the coal-handling system proceeded simultaneously, and this work was completed by September 1952. It was followed by the construction of the furnace, which was ready for tests in December 1952. The first quarter of 1953 was spent in testing the furnace. These tests are analyzed in Appendix 3, where a full description of the original furnace is also included.

This experimental work resulted in a new furnace design (described on pages 19-22 of this report) featuring a completely changed mixing section. By that time other components of the plant began to arrive from the subcontractors and the assembly of the complete test plant commenced.

Finally, by the end of October 1953, the plant was completed and ready for initial tests. It was motored on the starting engine on November 5 in order to check the overspeed governor blow-off valve, tachometers, etc. Another motoring run made on November 9 resulted in minor adjustments of the furnace dilution air-ports. The air meters were completely calibrated, and the plant was finally declared ready for full-scale tests.

TEST SERIES

• Tests with Dart No. 8 Engine (November 1953-April 1954)

The first test run was scheduled for November 16. As always in the case of a new and untried design, a careful and methodical procedure was evolved. An elaborate checklist containing more than 70 items and dealing with every detail of the experiment was prepared in order to insure that all components were operating properly (16). The starting procedure was also laid out in detail, but of necessity it had to remain flexible as it was impossible to predict accurately the exact temperatures and speed at which the plant would 'take-off'.

To make it possible to investigate fully the behavior of the plant while it was still at moderate temperatures and speed, the first test schedule called for several hours of 'assisted running', with the gasoline starting engine supplying the power deficiency. Kerosene was used as fuel in the cyclone combustor and temperatures of 1,020°F at furnace outlet and 800°F at turbine inlet [21] were reached at a speed of 7,260 rpm (17). Ignition and control of the furnace was found to be very easy, and valuable experience of the operation of the plant was obtained under safe conditions. Several minor leaks were discovered and corrected, and the results of this test were used in predicting more closely the conditions at which the plant would become self-supporting.

The first fully operational run on fuel-oil was scheduled for November 20, 1953.[22] The gas-turbine unit was rotated by the starting engine, and the furnace was lit on fuel-oil at a turbine speed of 3,000 rpm (23 per cent of full speed). The fuel flow was then gradually increased, and as the temperatures went up the turbine

[21] The two temperatures most representative of the plant performance are those at furnace outlet and turbine inlet. Henceforth in this report, when two temperatures are given without specific reference, they will always refer to the furnace outlet and turbine inlet temperatures, in that order.

[22] A complete history of the experimental work is given in Appendix 7.

supplied a progressively greater proportion of the total power required to drive the set. As the speed increased, the gasoline engine was gradually throttled down until it was producing no more power. It was then declutched, and the gas-turbine set accelerated under its own power.[23] The plant became self-sustained at 11:52 a.m. on November 20, 1953, at 1,345° and 1,060°F and 6,200 rpm. It was then run on oil under its own power for five hours at 10,000 rpm and temperatures of 1,560° and 1,065°F. The power developed varied up to 55 hp. The run showed that inadequate circulation of cooling air was causing a slight overheating of the hot-heat-exchanger shell, and action was taken to remedy this. Otherwise, the test was considered successful and the first run on coal was scheduled for November 27.

The same starting procedure was followed on November 27. The furnace was lit at 9:20 a.m. and at 10:56 the plant was fully operational on oil. Then gradually, over a period of five hours, the speed was increased to a maximum of 12,000 rpm, and the temperatures to 1,580° and 1,167°F. At 4:08 p.m. coal was substituted for oil [24] and the engine was operated on coal for the first time at about 11,500 rpm and at 1,470° and 1,110°F. The coal rate was about 630 lb/hr and the plant developed about 50 hp. The plant was run for two hours under these conditions and was then shut down.

This was the first successful run of an exhaust-heated coal-burning gas turbine in the world and represented a fitting climax to three years of intensive effort on the part of all engaged on this project. A full inspection of the plant was made after this initial run on coal. The furnace was found to be in good condition. The walls of the cyclone and the hot zone of the mixing section were covered with a fairly uniform layer of slag. The drain hole, however, was completely blocked by frozen slag. This was attributed at the time, to low-temperature operation, but in subsequent runs slag-drain blockage proved to be one of the most difficult problems to solve.

The run showed that further modifications to the hot-heat-exchanger cooling systems were required in order to provide a more uniform distribution of tube-plate and shell temperatures. The ceramic inserts at the inlet to the tubes were in excellent condition.

On December 4 another run was attempted, but after 15 minutes of operation on coal the feed became erratic and, to maintain operating conditions, oil had to be partially turned on again while attempts were made to correct the coal feeder. While this work was in progress, a mild explosion occurred and the bypass became overheated. The plant was stopped immediately.[25]

[23] While the gasoline engine was being throttled down it was periodically declutched for a fraction of a second. As long as it tended to acclerate when declutched it was still driving the gas-turbine set. When, finally, there was a tendency for the speed to drop upon declutching, the engine was left declutched. The gas-turbine set, now free of the burden of the gasoline engine, accelerated quickly.

[24] The change from oil to coal was effected very simply by opening the coal shut-off valve and progressively reducing the oil flow while the coal supply was increased until no oil was being used and the plant was operating entirely on coal.

[25] It is interesting to note that this proved to be the only emergency stop made necessary by immediate safety considerations during the whole series of tests.

Inspection revealed that a blockage of partially burned coal had occurred at the throat of the entry to the furnace. This caused additional coal to back up into the bypass ducting where it ignited, thus overheating the ducting and the cold heat exchanger. No serious damage occurred, except to the bypass duct, which had to be replaced (17). The time required to make and install a new duct was also utilized to have the subcontractor repair the expansion joint of the cold heat exchanger, which seized on both previous runs owing to faulty manufacture.

One of the major sources of trouble, which plagued the experimental work for nearly six months, arose from the gross mismatching in size of the light aircraft engine with the heavy industrial-type ducting to which it was coupled. The problem of connecting the thin, flexible components of the turbine to the heavy manifold and then keeping them in perfect alignment notwithstanding the thermal expansion and movement of parts in operation proved to be an engineering task of the first magnitude. If there was any malfunction of the slip joints with which the manifold connections were fitted, the load would be transmitted to the turbine casing, thus distorting it and causing a rub of the turbine blades. The rub never occurred while the engine was running, as the slip joints appeared to function properly when heating up. Several hours after a test, however, when the whole assembly had cooled down to room temperature, a rub would be found (18, 19, 20). Rectification of this trouble was time-consuming, as the turbine had to be completely disconnected from the ducting, and it was usually from two to four weeks before the next run could be made.

Many solutions were tried, and gradually, by completely redesigning the slip joints, suspending the turbine from the manifold ducting and providing an elaborate system of counterweights to keep the engine in alignment while running, these troubles were overcome (20). They would not, of course, have arisen had a properly designed industrial gas turbine been used, and they cannot be regarded as being in any way related to the problem of burning coal in a gas turbine.

The testing during this initial period was limited to speeds not exceeding 11,000 rpm. This was made necessary by the fact that when the rotating inlet guide vanes of the Dart compressor failed during the furnace tests, they were replaced without any dynamic re-balancing of the assembly (no facilities being readily available), and this resulted in severe vibration at higher speeds. Although a new and better engine (Dart 16) was available as a spare, it was thought best not to install it until the suspension difficulties had been resolved.

Most of the tests carried out during this period were of relatively short duration (the longest non-stop test lasted 13 hours, of which 9½ were on coal) and concerned mainly problems of combustion and slag removal.

Several different methods of injecting coal into the furnace were experimented with. The original method involved the use of a 6-inch primary-air duct (branching off the turbine exhaust duct), into which coal was injected and conveyed to the coal-entry section of the furnace (Figure 14), which the coal-air mixture entered tangentially to promote swirl. An additional air stream (tertiary) was brought in at this point.

It was when the entrance from this section into the furnace proper became blocked that the coal backed up the primary duct and into the bypass duct and caused the emergency stop already mentioned. To prevent the recurrence of this failure, the primary- and tertiary-air supplies were dispensed with, and coal was fed directly into the furnace by the coal-injector forcing air (18).[26]

Initially, the coal was injected axially into the furnace by a jet of compressed air. A conical target was placed in front of the injection pipe to spread the coal on the walls. This resulted in excellent combustion as long as the target was in place. However, no adequate means (such an insulating, water-cooling, etc.) of prolonging its life beyond a few hours could be found. Since axial discharge of coal without the target resulted in excessive carryover and heavy deposition of slag on the tube-plate inserts, this method had finally to be abandoned, although a considerable amount of running was done before other arrangements were tried. By the end of March 1954, after some 54 hours of operation at reduced speeds, the manifold-expansion difficulties were overcome and the Dart-16 engine was installed.

• Test of May 19-21, 1954

This first test with the new engine, which in many ways represented a milestone in the operation of the plant, was continued non-stop for 57 hours.

Axial discharge into the furnace was used in spite of the knowledge that combustion would not be good, but the installation of the larger Dart engine made conditions even worse than expected.

With power outputs of up to 250 hp. obtainable from the new turbine, a coal-feed rate of 900 lb/hr was measured as compared with the previous rate of about 500 lb/hr (21). At this higher rate, the coal injector required more than twice the amount of air for a given quantity of coal, so that the injection velocity increased to nearly four times its original value.

As a result the coal particles were thrown farther downstream before striking the walls. This meant that at times most of the combustion was occurring in the mixing zone and leaving a heavy deposit on the heat-exchanger tube plate. This, of course, was not known at the time, and the run was continued until the deposit became so heavy that the heat-exchanger tube-temperature readings became erratic and the performance of the whole plant was seriously affected.

Another fact that added to the severity of operating conditions was the early blocking of the slag hole and the consequent stoppage of all slag drainage after the first 90 minutes. In spite of this, it was decided to continue the test. Sight ports

[26] This arrangement was first tested in Ottawa in the Mines Branch furnace and was found quite satisfactory.

in the furnace made it possible to observe the slag level, which was allowed to mount gradually until there was a foot of solidified slag at the bottom of the mixing section.

When the run was finally terminated and the furnace opened up, the reason for the erratic tube-temperature readings became clear. As shown in Figures 22 and 23, the slag accumulation in the mixing zone was very heavy and the effective area of flow through the tubes was reduced to about 55 per cent of the clean value by the slag deposit on the tube plate.[27]

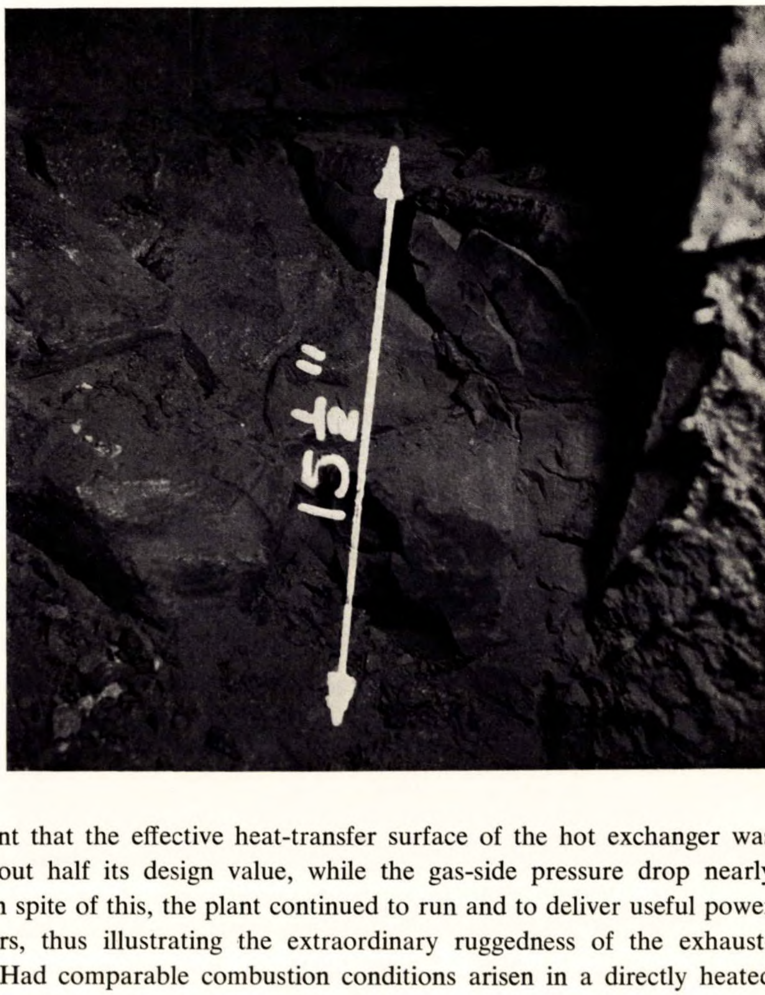

Figure 22. Mixing section after run of May 19.

This meant that the effective heat-transfer surface of the hot exchanger was reduced to about half its design value, while the gas-side pressure drop nearly quadrupled. In spite of this, the plant continued to run and to deliver useful power for many hours, thus illustrating the extraordinary ruggedness of the exhaust-heated cycle. Had comparable combustion conditions arisen in a directly heated cycle the test would probably have ended in disaster.

[27] A detailed analysis of the blockage is presented in the Analysis of Results section, on page 54 of this report.

As it was, the heat exchanger was undoubtedly saved by the ceramic inserts protecting the tube plate. Most of the slag was deposited on them and, although most of the inserts had to be replaced, the tubes appeared clean and undamaged. There were a few droplets of slag in some of the tubes just beyond the length occupied by the inserts, but they were easily dislodged. Outwardly there was no appearance of damage, but subsequently it became clear that the slight deposit of slag inside the tubes that were almost completely blocked at the inlet could have produced conditions favorable to the initiation of a corrosive attack of sulphur on the tube metal. (Actually, the first corrosion failure did not occur until some 140 hours of further running.)

Figure 23.
Tube plate after run of May 19.

Analysis of the fuel consumption that occurred during the test reveals that out of some 56 hours of testing nearly 16 hours were run on oil. This was due to the small capacity of the crushed-coal hopper, which needed refilling after some five to nine hours of running. Although coal could be processed while the plant was running, it was found that the refilling of the hopper prevented steady metering of the coal so that the plant had to operate on oil during that time. To make steady operation on coal possible, an additional hopper of 2,000 pounds' capacity was installed after the test, in which the processed coal could be stored and from which it could be dumped into the scale hopper in a few seconds.

Another modification to the coal-handling system consisted in pressurizing the scale hopper up to about 20 inches of water pressure. This was done because progressive blocking of the heat-exchanger tube-sheet caused a continuous change in the furnace back pressure, thus adversely affecting the metering of coal. With increased pressure in the hopper, the coal discharge was less affected by changes in furnace pressure.

All other major components of the plant, such as the Dart engine and the cold heat exchanger, operated satisfactorily and there was no further trouble with the engine suspension.

• Tests of July and August, 1954

The six tests run during this period were characterized by an almost complete absence of mechanical troubles. Consequently, all development work was concentrated on the real problems at hand, i.e. efficient combustion, slag-draining and tube-plate deposition. Several types of coal entries were investigated. To prevent axial discharge into the furnace, the coal-injector pipe was inclined at 30 degrees to the centre line (test run of July 5 and 6). This resulted in good combustion as long as the coal feed remained steady, but this unfortunately was not the prevailing condition. Irregularities in the flow would cause the flame front to move periodically out of the cyclone section and into the mixing zone, where combustion would occur directly in front of the heat exchanger and result in excessively high temperatures at the tube plate. Under these conditions, the slag deposition on the inserts was heavy, and the test was discontinued after 18 hours.

To secure good combustion in the cyclone zone, various adjustments in the size of the air-inlet ports in the furnace were tried on subsequent runs (22) and some improvement was obtained. The slag-hole problem remained unsolved, however, and most of the tests were terminated owing to unsatisfactory drainage.

• Tests from September 1954 to April 1955

Seventeen tests were run during this period. They were usually of relatively short duration (less than 10 hours),[28] and concerned mainly the investigation of the effects of gas temperature and combustion performance on the tube-plate deposition. Temperature traverses were carried out at the heat-exchanger inlet to obtain a correlation between temperature and rate of deposition (Figure 33, page 55.)

[28] The only exception was the test of December 14-15, which lasted more than 24 hours and was run for 21 hours, 43 minutes, on coal.

In order to investigate the composition of the fly-ash in the gas stream, a special sampling probe was designed which could be traversed across the heat-exchanger inlet duct so that a representative sample of the fly-ash could be collected and analyzed.

Experimentation continued with different types of slag holes, but it became increasingly evident that a complete relocation of the drain hole would be required to secure good drainage. Efforts to keep the slag molten and flowing by means of thermal or electrical heating or by mechanical methods proved ineffective, and nine of the 17 tests were terminated because of inadequate drainage.

Figure 24. Re-entrant throat with slag-draining cut-out (viewed from inside cyclone).

In December 1954, in an effort to improve combustion, a re-entrant throat was installed. An insulated water-cooled steel pipe was used (Figure 24) as a makeshift arrangement so that the throat could be installed quickly with the furnace in position. A marked improvement in combustion was noted (as exemplified in the 24-hour run on December 14) as long as the throat was in good condition, but a gradual erosion of the insulation invariably took place and resulted, with time, in a steady deterioration of performance. However, this had to be endured; and until a new furnace with a permanent throat could be built, repairs had to be made before each run.

Experiments with various coal entries showed that combustion conditions were very sensitive to minor alterations. Even when good combustion was obtained with a clear entry, conditions deteriorated gradually as slag and clinker deposits formed around the discharge pipe. When compressed air was used as a fluidizing medium for conveying the coal to the furnace, factors such as furnace back-pressure and even the moisture in the coal affected the steadiness of the discharge. To avoid these difficulties it was decided to try feeding the coal into the furnace with a feed-screw.

An additional coal hopper was installed just outside the test cell with a short run of straight pipe discharging directly into the furnace. A 2-inch screw conveyed the coal from the hopper to the furnace, the rate of flow being controlled by the speed of the screw. Various arrangements such as swirl vanes, air jets and conical entries were tried at the discharge end, but none of these gave completely satisfactory results.

As a result of these tests, however, often by a process of long and tedious analysis and elimination, there gradually emerged a picture of what a successful design should look like, what features of the original furnace it should retain and where drastic modifications should be made. These changes were incorporated in the new furnace, built for the second series of tests, which began early in 1956.

• Tube Corrosion

Following the run of December 28, 1954, after some 196 hours of operation on coal and a further 69½ hours on oil, 48 tubes were found to have developed leaks. Investigation revealed that these tubes were corroded to a varying degree. Some had only a few small holes, but others had been attacked more extensively. One had been virtually cut off a short distance from the hot tube plate.

Six tubes were removed for detailed inspection and metallurgical analysis, and their tube-sheet openings, together with those of the remaining 42, were plugged before the next test. Thus the heat-transfer surface of the exchanger was reduced by nearly 10 per cent.

Six additional tubes were plugged on February 8, 17 on March 18 and another 73 on April 13. Thus the tubes affected in 314 hours of running totalled 144.[29] Since operation without these tubes would have reduced the heat-transfer surface by nearly 30 per cent while increasing the gas-side pressure drop by about 60 per cent, it was decided to interrupt the experimental running and to rebuild the heater and the furnace, the new design being based on lessons learned in this first series of tests.

The analysis of the test results which led to the evolution of new designs is presented in the pages immediately following.

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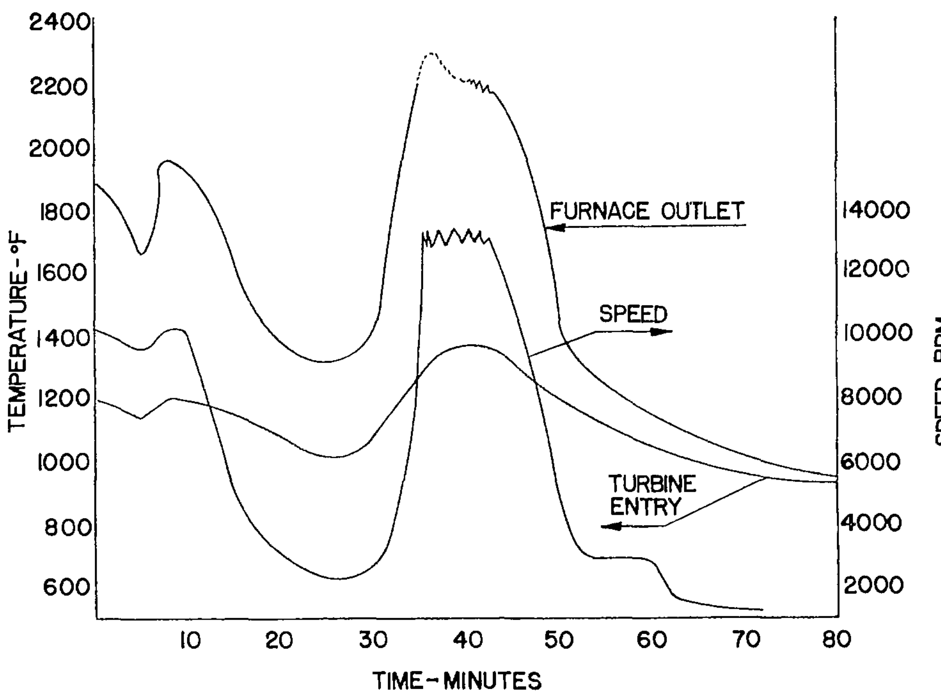
[29] A detailed analysis of the corrosion problem begins on page 94.

ANALYSIS OF RESULTS

• Over-all Performance of Plant

1. Operating characteristics, control

Two of the most outstanding features of the early operation of this plant were its ruggedness under adverse operating conditions and the quick response to variations in fuel flow. When the cycle was originally conceived, it was expected that the large thermal capacity of the heaters and furnace would result in a somewhat sluggish response, so that quick control would have to be effected by regulating the bypass flow. As it turned out, it was never necessary to use the bypass flow in this manner, and the coal rate was the only control used. Changes in the power output of the plant were extremely rapid after a change in the fuel-supply rate, and were more rapid than the response of the pyrometry equipment, which was by no means slow. By control of the fuel supply only, full power (250 hp) could



Time-Minutes		Time-Minutes	
0 - 7	Unsteady coal feed	37 - 42	Governor blow-off operating
10	Coal valve control and injector air shut off	35 - 40	Temperature off chart
20 - 30	Coal valve failed to operate and coal flowing to the furnace at low speed	42 - 53	Coal now burned out and plant decelerating
30 - 37	Manual emergency valve closed but plant accelerating	53	Starter motor engaged for motoring

Figure 25. Governor operation.

be thrown off or restored in about equal intervals of 15 seconds. It could be dropped instantaneously by use of the overspeed governor and blow-off valve,[30] but this method was not normally used.

On one occasion, however, the governor undoubtedly saved the plant from serious damage. The plant was being stopped to investigate coal-feed troubles and the coal shut-off valve control had been operated. The first symptom of trouble

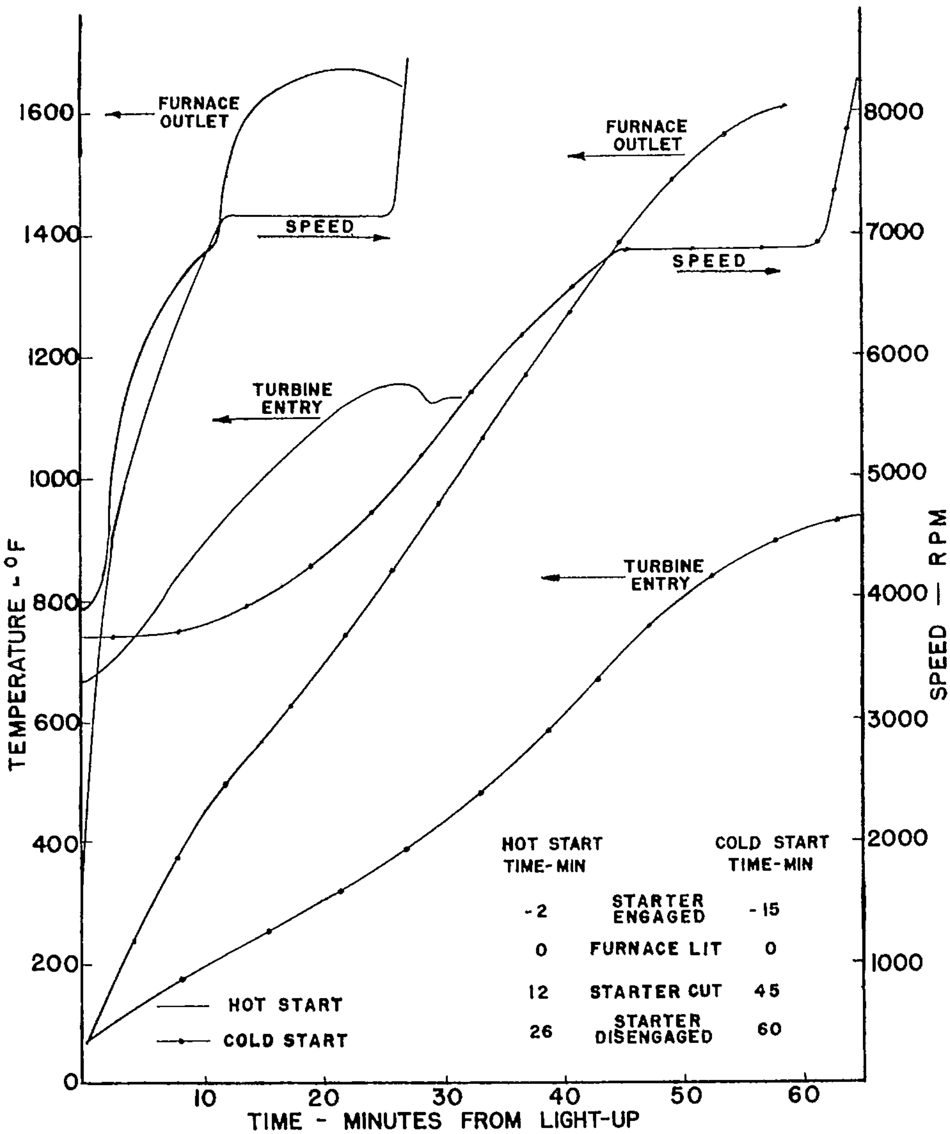


Figure 26. Starting cycle.

[30] See page 29 for a description of the overspeed governor.

was that the plant was taking a very long time to stop. After slowing to about 2,000 rpm it continued to run at this speed. It was then realized that the coal shut-off valve had not closed and that coal was flowing into the furnace from the hopper under the influence of hopper pressure alone, which was higher than furnace pressure at the low speed of the plant. The coal supply was then shut off with an emergency manual valve, but the plant was now accelerating at very high temperature resulting from combustion of the large quantity of coal with which the furnace was largely filled.

The operator was on the horns of a dilemma: either he could open the blow-off valve and stop the plant, thus running the risk of an explosion from the gases that would be distilled from the large quantity of unburned coal in the furnace, and producing very high temperatures; or he could allow the plant to run up to high speed and burn out the coal as quickly as possible. He chose the latter alternative. The plant accelerated to maximum speed with heater-inlet temperatures in excess of 2,200°F, and ran on the governor at this speed for about three minutes. Then the temperatures subsided and the plant slowed down. Figure 25 is a plot made from the recorder chart illustrating the sequence of events. No damage could be found after this incident, and the plant was restarted 45 minutes later for a 7-hour test run.

2. Starting and stopping

The starting procedure adopted for the plant is described on page 32. The plant was usually brought up to temperature fairly slowly through a gradual increase in the fuel-oil flow while the gas-turbine set was still being assisted by the starting engine. The purpose was to reduce the thermal shock sustained by the ceramic materials of the furnace when the plant was started from cold. Figure 26 shows that it took about 60 minutes for the plant to become fully self-supporting. The maximum power used during starting was about 25 hp, but the plant could be started on as little as 10 hp if the need arose.

When hot after a short stop, the plant has been restarted from stationary in 15 minutes. Figure 26 shows a slightly slower start, the complete disengagement being effected at the 25th minute.

Stopping the plant from full temperature in a few minutes involved no difficulty or danger. This was done on many occasions so that a change could be made for test purposes or for adjustments. The various cooling-water flows were maintained in the plant and no significant increase of any temperature occurred. After most tests it was necessary to inspect the inside of the furnace or heater as soon as possible. To reduce the cooling time, the plant was motored by the gasoline engine for several hours after the flame was extinguished. Figure 27 illustrates a typical stop with and without subsequent motoring (23).

Another important point relating to the operation of the unit is that, once the initial mechanical troubles were overcome, all running procedures were reduced to routine operations regularly performed by a crew whose only previous experience with power-generating machinery was limited to the automobile engine.

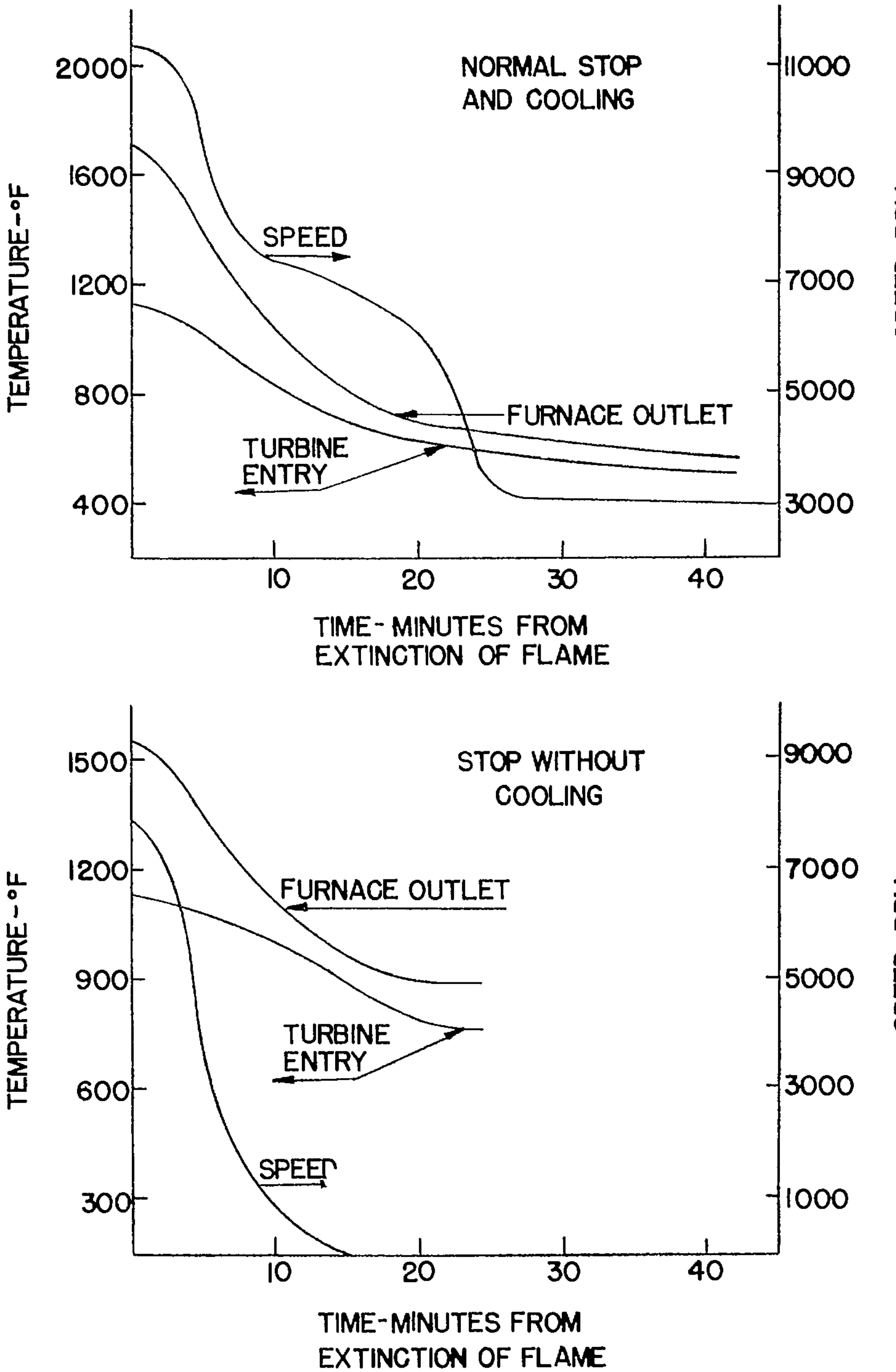


Figure 27. Stopping cycle.

3. Power output, efficiency

In calculating the thermodynamic cycle on the basis of the Dart gas-turbine characteristics, it was established that the maximum power output to be expected at design conditions (13,000 rpm and 1,340°F turbine-inlet temperature) was about 500 hp and that this decreased to about 400 hp at 11,500 rpm (Figure 28). The maximum power obtained with the Dart 16 during the first test series was 290 hp, and in a number of tests run at 11,500 rpm the engine produced 260 hp. A detailed analysis of the test results revealed that the low output was primarily due to an excessive pressure drop on the shell side of the hot heat exchanger. This is considered more fully under heat exchanger performance (pages 50-54).

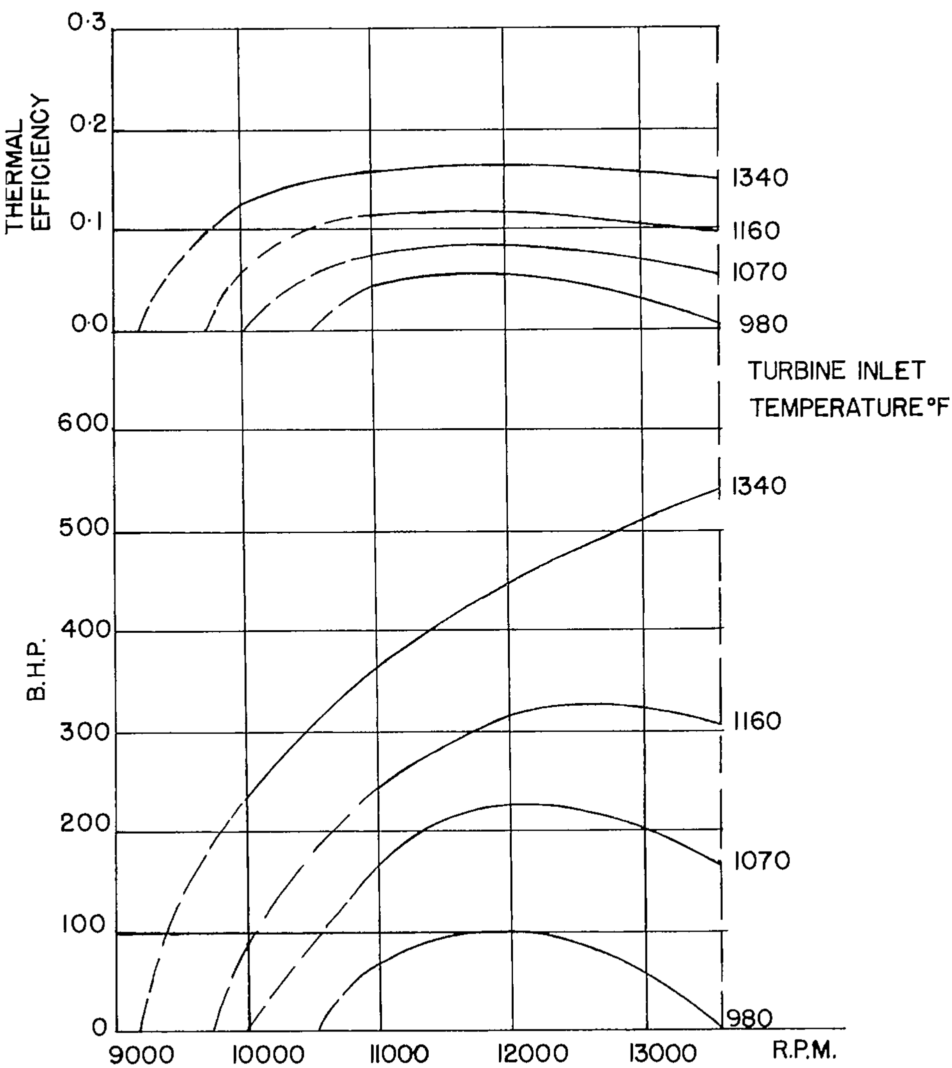


Figure 28. Estimated performance of coal-burning turbine.

With this excessive pressure drop a temperature of about 1,410°F was required at the inlet to the turbine (against a 1,340°F design limit) in order to produce 260 hp. This in turn meant that the maximum gas temperature (at the inlet to the hot heat exchanger) was between 1,830° and 2,000°F (1,000 to 1,100°C). Although not excessive from the point of view of thermal stresses, this temperature resulted in accelerated fouling of the tube plate. Hence most of the running was done below 1,830°F (1,000°C), and the power outputs were correspondingly reduced.

This in no way detracted from the value of the tests, as the primary object of the experimental work was the testing of such components as the heater and furnace, and not the production of power.

The excessive pressure drop was also partly the cause of the somewhat low thermal efficiencies obtained. It must be borne in mind that the thermal efficiency is composed of three factors:

(*a*) thermodynamic-cycle efficiency
(*b*) combustion or heating efficiency
(*c*) mechanical efficiency.

In order to calculate the cycle efficiency, factors (*b*) and (*c*) must be found. The mechanical efficiency can be assumed to be close to 100 per cent for this type of engine, and the combustion efficiency can be obtained from a heat balance on the furnace and the heat exchanger. The thermal efficiency is measured directly as the ratio of power output to fuel energy supplied, and thus the cycle efficiency can be obtained:

$$\text{Cycle efficiency} = \frac{\text{Thermal efficiency}}{\text{Heating efficiency} \times \text{Mechanical efficiency}}$$

This efficiency includes all losses in the turbine set and the effects of the heat-exchanger pressure losses and, in fact, corresponds to the over-all efficiency of a conventional set, divided by its combustion efficiency. A plot of the cycle efficiency thus obtained is shown in Figure 29. With a turbine-inlet temperature of 1,292°F, a maximum value of nearly 14 per cent was obtained. The sensitiveness of the efficiency to the turbine-inlet temperature is demonstrated by the fact that a 90°F drop in it causes a 40-per-cent reduction in efficiency.

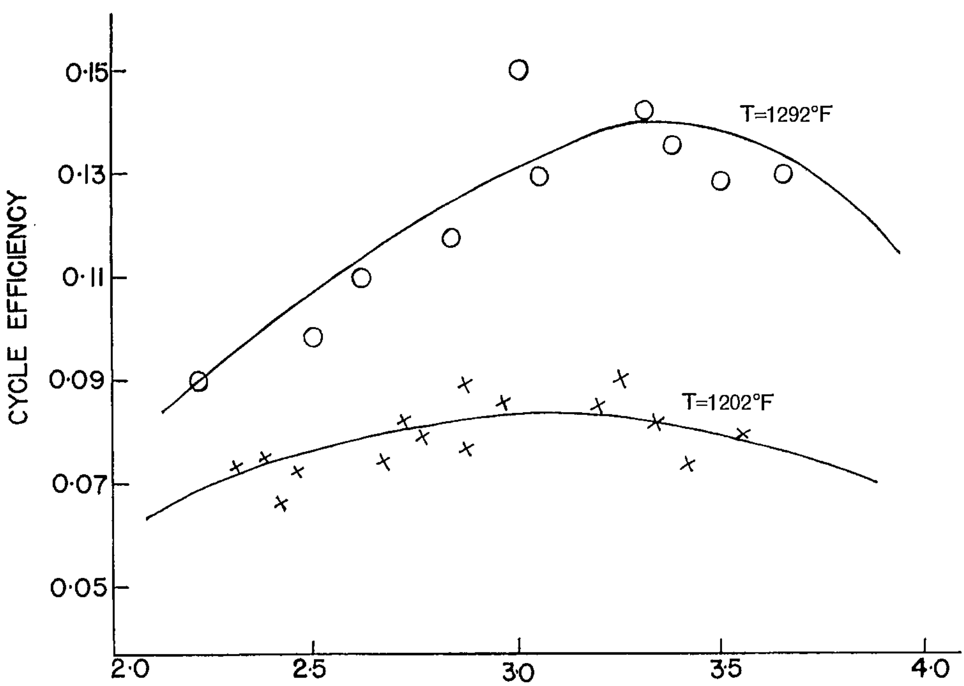
4. Heating ratio

The heating efficiency that is significant in this cycle is not the same as the combustion efficiency of a conventional cycle. A better name for it would be *heating ratio,* and it should be defined as the heat actually supplied to the compressed air in the hot heat exchanger divided by the heat input to the furnace. If the surface of the heater is sufficiently large to reduce the gas temperature below the turbine-exhaust temperature, the heating ratio could exceed unity. In the split cycle used, the heater was designed to reduce the gas temperature to the same value as the temperature of the bypass air (or turbine exhaust) so that the maximum value of the heating ratio would be one. The factors which prevent the attainment of this value are as follows:

(*a*) unburnt combustible in gas or ash
(*b*) furnace cooling-water losses
(*c*) tube-plate cooling-water losses
(*d*) heat losses by radiation and convection from heated surfaces.

A considerable effort was made to measure the aforementioned losses, and the results of this work are discussed below.

(a) Unburnt combustible in gas or ash—No effort whatever was made to burn a uniform grade of coal, and the ash and moisture content varied considerably throughout the test program.



T = Turbine entry temperature

Figure 29. Cycle efficiency.

Table II shows extreme variations in the ash and moisture and sulphur content of the coals used and the approximate average values during the period under analysis (on the basis of Reference 28a).

Table II

Ash, Moisture and Sulphur Content of Coal

—	Ash Content	Moisture	Sulphur
Maximum	21.1%	2.0%	5.3%
Minimum	4.3%	0.4%	1.7%
Representative average	11.0%	0.8%	3.3%

In order to determine the losses due to unburnt combustibles the slag was collected carefully after each test and samples were sent for analysis to the Fuels Division, Mines Branch, Department of Mines and Technical Surveys, in Ottawa.

The composition of the slag varied somewhat with the location from which it was obtained. The major part was collected from the slag box (provided it was operating properly), but considerable amounts were sometimes also removed from the furnace mixing zone, the heater inlet and the ash trap near the exhaust stack.

The slag collected from the slag box usually showed from 98 to 100 per cent ash, except during the early 1954 runs when combustion was very poor, while the samples removed from the heater inlet contained up to 10 per cent combustible.

From 5 to 30 per cent of the total weight of the ash collected was in the form of fly-ash. Most of this was collected in a special ash separator constructed at the inlet to the exhaust stack or in the exhaust stack itself. Samples were also obtained by traversing a sampling probe across the furnace outlet duct during the test.

The carbon content of the fly-ash was invariably much higher than that of the slag, averaging from 60 to 70 per cent by weight.

From these measurements it was possible to estimate the losses in unburnt combustible and also to calculate the sensible heat loss in the slag.

(*b*) *Furnace cooling-water losses*—In order to reduce the temperature of the metal walls and increase the life of the refractory linings of the furnace, the walls of the cyclone and the baffles were water-cooled. This resulted in a considerable loss of heat, which in the cycle under test was not recoverable. The only way to reduce this loss with that particular design of furnace was to increase the thickness of the refractory lining. This, however, even with the best materials available, proved impossible, as the molten slag gradually fluxed away most of the refractory until, after about 100 hours of operation, the lining consisted almost entirely of slag. The thin wall did not, of course, prevent good combustion, the chrome-ore lining being required only during the initial starting period as a 'wettable' surface for the slag; but it did result in a high cooling-water loss, which amounted to 15 per cent of the heat supplied by fuel. This, of course, would not constitute a loss in a cycle in which steam could be employed usefully.

(*c*) *Tube-plate cooling-water losses*—A similar loss occurred in the hot heat exchanger, where cooling water was used to reduce the temperature of the hot-end tube plate. Again the loss was severe, amounting to 9 per cent of the heat input. The results obtained with the heat exchanger (and discussed later in this report) suggested that the use of an air-tube heater would be perfectly feasible. This would permit operation without a water-cooled tube-sheet and consequent improvement in efficiency.

(*d*) *Heat losses from heated surfaces*—These losses were computed by taking surface-temperature measurements and estimating heat-loss coefficients and surface areas. They were checked by installing a carefully calibrated extractor fan in the test cell and measuring the total enthalpy increase of the air passed through the cell. This heat loss was apportioned to the individual components, as already indicated.

Table III shows a heat balance for a typical test.

Table III

Distribution of Heat Losses

Heat supply in fuel	100%
Furnace losses:	
(a) Unburned combustible and heat in slag	4%
(b) Furnace cooling water	15%
(c) Heat loss from casings	7%
Total furnace losses	26%
Hot heat exchanger:	
(a) Tube-plate cooling loss	9%
(b) Heat loss from casings	3%
Cold heat exchanger:	
(a) Heat loss from casings	1%
Manifold and ductings	1%
Total plant losses	40%
Effective heating efficiency	60%

An examination of the foregoing table shows that, apart from the large cooling-water loss, a high proportion (12 per cent) of the heat input is lost by convection and radiation from exposed surfaces. This, of course, was an inevitable penalty for adopting a layout that would allow ready access to, and ease of dismantling of, all major components. It must not be regarded, therefore, as a particular characteristic of this type of engine.

• Performance of Individual Components

1. Gas-turbine set

The chief advantage claimed for the exhaust-heated cycle was that, by virtue of running on pure air, the gas-turbine set should be completely free from trouble and require only the most elementary maintenance. This judgment was completely vindicated: once the early suspension troubles were overcome, the only maintenance the turbine set required was limited to adding lubricating oil and an occasional washing of the compressor with kerosene.

2. Hot heat exchanger

(*a*) *Thermal performance*—As already mentioned, the heat-transfer coefficients were calculated by using tube-flow correlations for both the tube and the shell side of the heat exchanger (Appendix 2).

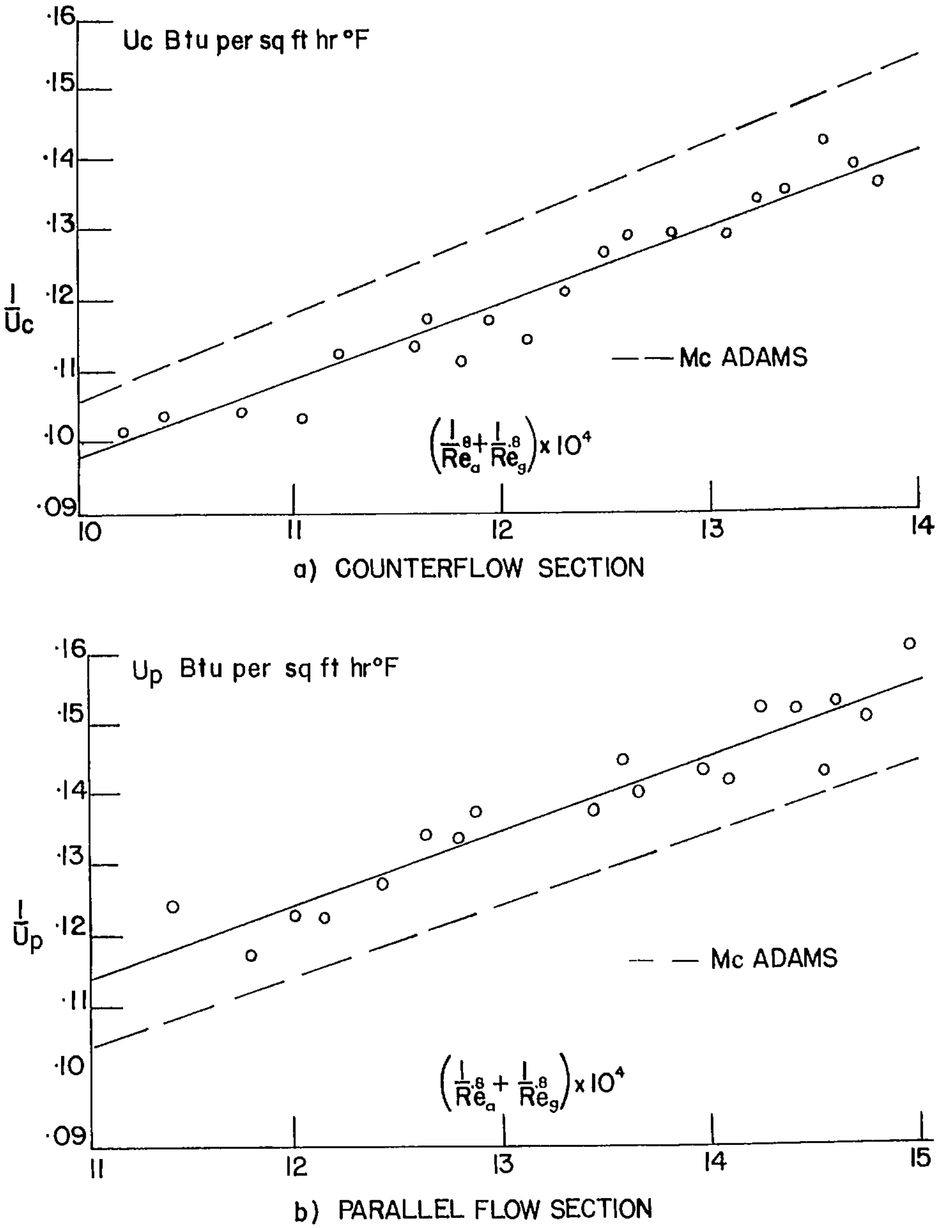


Figure 30. Hot-heat-exchanger over-all coefficient vs. flow parameter.

Since these correlations do not take into account the additional turbulence produced by the shell baffles, it was expected that the shell-side coefficient would be higher than predicted. The over-all increase in performance would not be very pronounced, since the shell-side coefficient is not controlling. Better-than-design performance, however, was expected, and test results show that it was obtained in the counterflow section. Representative results of a number of tests are plotted in Figure 30(a), in which $\frac{1}{U_c}$ the reciprocal of the over-all coefficient (i.e. the over-all thermal resistance), is plotted against a parameter representing the effect of individual film resistances.[31] The measured coefficient U_c is about 10 per cent higher than the calculated value.

Since the parallel-flow section of the heater was even more closely baffled than the counterflow, a similar effect would be expected there. This, however, was not the case, as is shown by Figure 30(b). The reason for this apparent anomaly lies in the fact that the temperatures used in calculating the over-all coefficient U are not truly representative of the thermal conditions near the hot tube plate. For instance, the gas-inlet temperature, which is measured in front of the tube plate, undergoes a considerable change before the tube plate is passed. As shown already in Table III, about 9 per cent of the heat added in the furnace is carried away by the tube-plate cooling water. This means that the effective inlet gas temperature available for heat transfer is much lower than indicated by the T-50 thermocouples. Similarly, the final temperature of the air on the shell side is reduced somewhat by the main tube-plate (rear surface) cooling air, which enters at a temperature of about 250°F. Finally, the air temperature at the transfer duct, from the counterflow to the parallel-flow section inside the exchanger, was originally measured by four thermocouples. Towards the end of the test series, however, only two of these were working properly and, as their readings diverged considerably at times, they were of limited value. All these temperature inaccuracies affect the parallel-section calculations considerably. The larger scatter of points in Figure 30(b) is evidence of this.

A good indication of the satisfactory over-all performance of the exchanger (in spite of the heat losses at the hot end) can be obtained by comparing the outlet-gas temperature with the bypass-stream temperature. The hot heat exchanger was designed to reduce the gas temperature to the same value as that of the bypass stream, and most of the test readings showed that these two temperatures were within 10 to 20°F. The design over-all thermal ratio of the exchanger (based on inlet and outlet temperatures) was 0.5 and, in spite of the cooling losses, this value was usually closely approached. Since the tube-plate cooling loss was measured, it was possible to calculate the gas temperature just inside the tube plate and thus obtain a corrected value for the thermal ratio. Both the corrected and the uncorrected values are plotted in Figure 31. In the particular test shown in Figure 31, trouble was encountered in flushing the slag, and two short stops were

[31] The advantages of this method of plotting are explained in detail in Appendix 8.

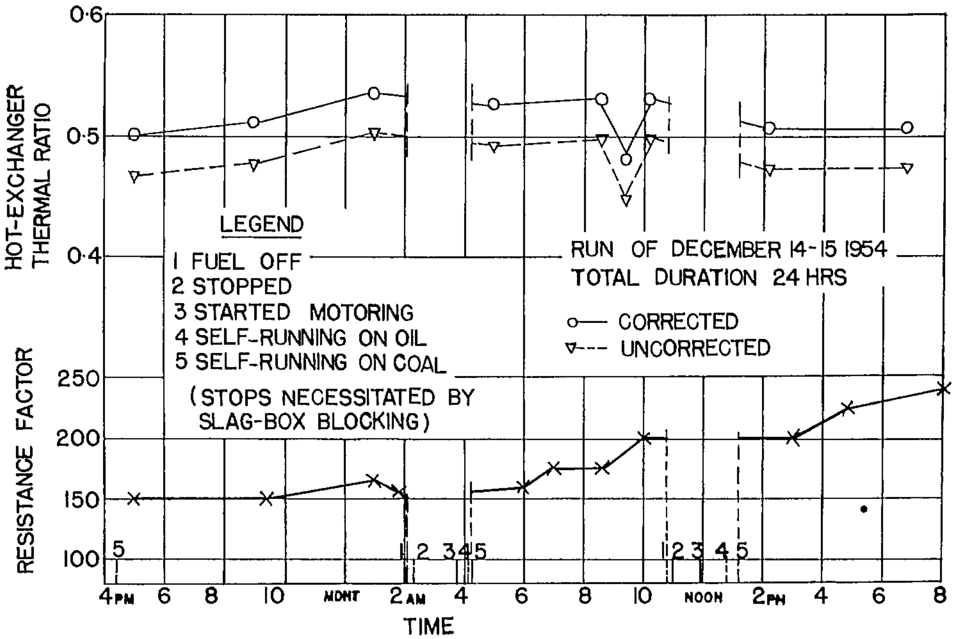


Figure 31. Performance of hot heat exchanger.

made to clean the slag out of the slag box. In spite of a steady increase of the gas pressure drop in the exchanger (indicating slag deposition on the tube plate), the heat-exchanger thermal ratio was not affected and, except for local variations, remained slightly above the design value throughout the test.

(*b*) *Pressure drop*—The tube-side pressure drops measured in actual tests and expressed as percentage losses are plotted in Figure 32(a). The abscissa represents a quasi-dimensionless flow parameter which takes into account pressure and temperature variations between different tests. A mean line through the points represents the average pressure drop with the tubes in a relatively (though not completely) clean condition. The design curve is seen to agree very closely with measured values. In the test in which a serious slag deposit occurred on the tube plate and blocked off a large percentage of the flow area, the gas-side pressure drop showed a gradual increase with time. By correlating the increase of pressure drop with the degree of blockage, a resistance factor was evolved and used as a criterion in determining the duration of trials in which gradual blockage of the tube-sheet occurred. This is discussed more extensively under fouling (page 54).

As already mentioned, the pressure losses on the shell side considerably exceeded the design values. This was due primarily to the effect of the tube support plates and to the counter-parallel flow transition section. References consulted at the time of the design (10) indicated that, if 'half-circle' support plates were used and placed as far apart as possible, consistently with their function of properly

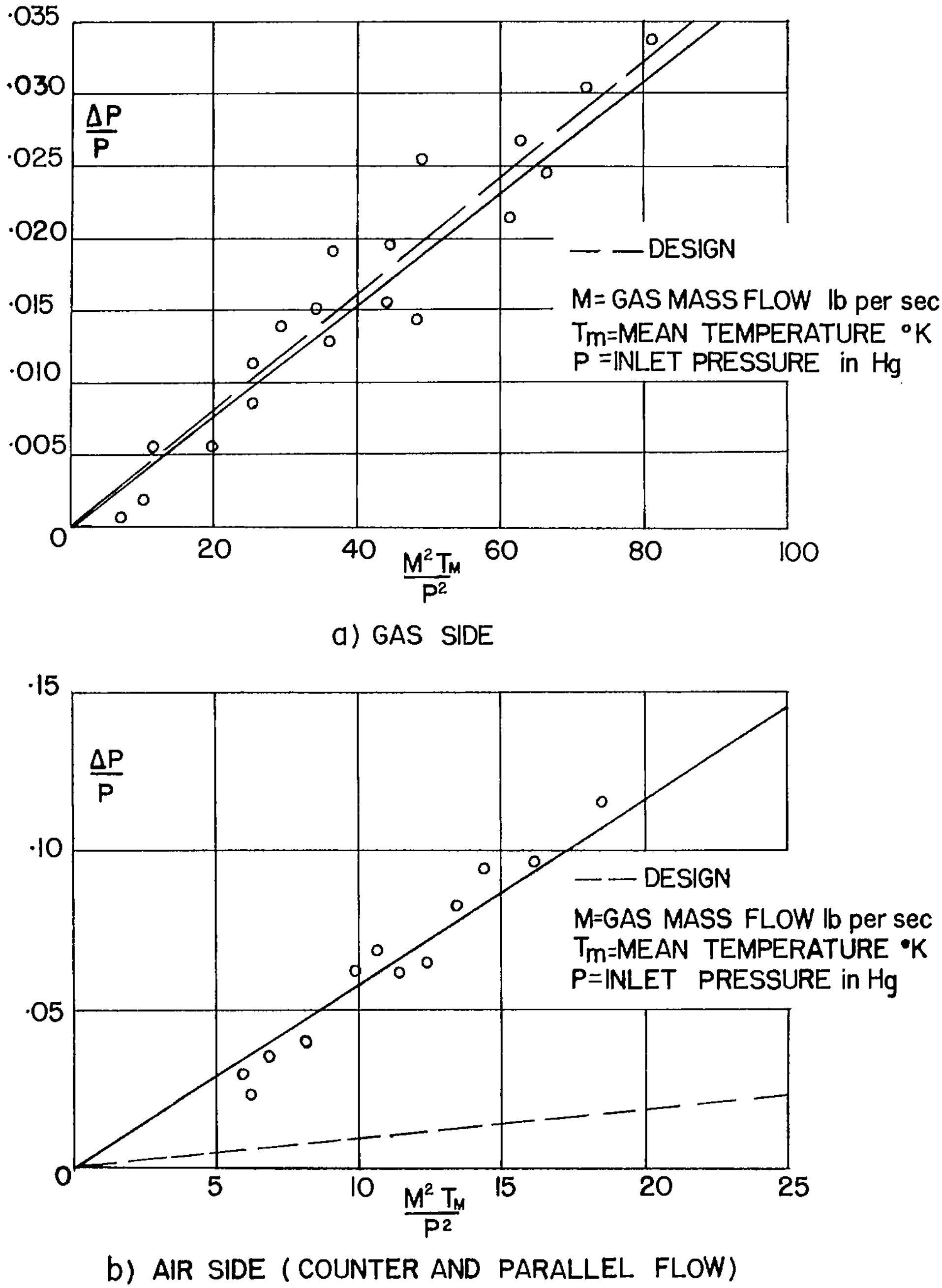


Figure 32. Pressure drop in hot heat exchanger.

supporting the tubes, the pressure drop could be computed by the use of longitudinal-flow correlations. In spite of this and in order to make some allowance for the baffle effects, the design pressure drop quoted was increased by about 30 per cent over the calculated value. Even this precaution, however, proved totally inadequate, and the actual pressure drop exceeded the design value by a wide margin (Figure 32b). The effect of this excessive pressure drop on the cycle performance has already been discussed.

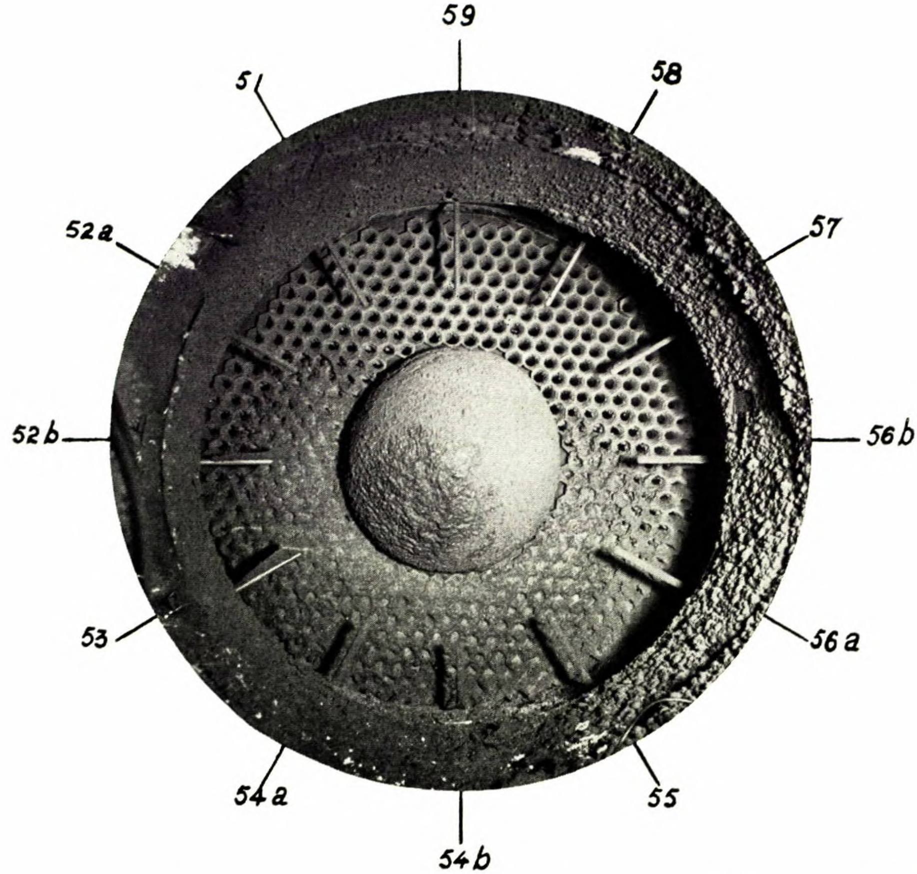
A detailed and exhaustive theoretical study of the flow on the shell side of the hot exchanger was made (24) in order to establish a rational basis for design, but this proved an almost impossible task. Purely empirical assumptions which could not possibly be justified theoretically had to be made for the sake of agreement with measured values. Thus, when the first test series was terminated and the rebuilding of the hot exchanger undertaken, a sectional small-scale model was built and extensively flow-tested in order to obtain a baffle configuration yielding the lowest pressure drop possible.

3. Fouling

(*a*) *Dry fly-ash*—One of the primary reasons for choosing a gas-tube arrangement for the heater was the fear of loss of performance by fouling of the tubes and hence the need for easy access to them for cleaning. While the heater was under construction, tests conducted at the Mines Branch on a small-scale exchanger consisting of only a few tubes suggested that dry-ash fouling of the tubes would not affect the coefficients. This was fully confirmed by the full-scale tests. Actually, there appeared to be no tendency whatever for the dry fly-ash, which had the appearance of a fine powder, to adhere to the tubes, the gas velocity being high enough to blow it right through. This ash usually collected in regions of low velocity and could be removed without difficulty. It originated mostly from the combustion of coal fines in the air stream and never presented an operational or maintenance problem.

(*b*) *Wet-ash or slag fouling*—This was the type of fouling which created a major problem in the operation of the plant. It consisted of deposits, on the tube plate and on the protecting screens in front of it, of slag that was still in a semi-molten and sticky condition and that adhered strongly to the surface on which it was deposited. Considerable testing time was devoted to the study of the mechanism and the reasons for this wet-slag fouling, and the following conclusions were drawn.

There appeared to be two distinct types of deposition. One occurred during periods of relatively good cyclone operation, while the other was definitely a result of improper combustion. The first type of deposit can be correlated very closely with the gas temperature prevailing at the heat-exchanger inlet. It is due to operation at gas temperatures approaching, or exceeding, the fusion temperature of the slag. Under these conditions, small particles of fly-ash carried by the gas stream remain in a soft and sticky semi-molten state until they strike the tube plate and literally freeze into slag. If the temperature of the gas stream is not



Position	1,880°F (1,025°C)	1,920°F (1,050°C)	1,960°F (1,070°C)	2,010°F (1,100°C)
58	2	—	—	—
59	3	—	—	—
57	4	3	—	—
51	4	3	1	1
52a	5	3	1	—
56b	6	3	2	2
52b	7	7	2	2
56a	12	9	6	4
55	13	9	8	6
53	14	11	4	4
54b	14	11	8	8
54a	14	12	9	7

Number of readings recorded at or above the indicated temperatures (readings taken at 30-minute intervals).

Figure 33. Correlation of temperature and degree of blocking of tube plate.

uniform, it can be expected that in the lower-temperature regions, where the ash is cooled to below its softening temperature, deposition would not be so heavy. This was confirmed by the run on July 5 and 6, 1954. Although combustion in this test was generally good, with relatively little carryover, the coal feed was unsteady and considerable fluctuations in the instantaneous value of the air-fuel ratio occurred. Satisfactory operation of the furnace depended on precise control of this air-fuel ratio between chemically correct and 20-per-cent-excess air. A greater excess of air reduced operating temperatures in the cyclone, causing greatly increased carryover and difficulties with slag removal. With less air than chemically correct the cyclone operated as a gas producer, discharging combustible gases into the mixing section. These gases burned in the dilution air, but as this was not introduced until after the baffles, combustion was still proceeding up to the heat-exchanger tube plate.

This resulted in high temperatures at the heat-exchanger inlet and subsequent rapid blocking of the tube plate. (The resistance factor doubled in about 12 hours of operation.)

Figure 33 illustrates the relationship between temperature and degree of blocking. The top region of the tube plate, in which the temperature did not exceed 1,880°F, was found completely clean, while the lower parts, which were often exposed to temperatures in excess of 2,000°F, were completely blocked.

An estimate of the blockage of the flow area that occurred in this run is given in Table IV.

Table IV

Degree of Blockage of Heat-exchanger Flow Area Run of July 5, 1958

—	Percentage of Total	Percentage of Through Area Remaining
Tubes completely open	33%	33.0%
Tubes 1/4 blocked	11%	8.3%
Tubes 1/2 blocked	11%	5.5%
Tubes 3/4 blocked	11%	2.8%
Approximate flow area remaining		49.6%

As mentioned on page 36, it is notable that the plant continued operating with such a drastic reduction in the effective flow area and heat transfer surface.

Another revealing conclusion that can be drawn from Figure 33 concerns the effect of operating temperature on tube-plate deposition and its relation to the initial fusion temperature of the slag. The fusion temperature varied from 1,870°F to 2,100°F for samples collected from the tube plate (28a). Figure 33 shows practically no deposits on inserts located in the region where the gas temperature never exceeded 1,900°F. From the foregoing it would appear that no trouble would be experienced if the heater-inlet temperature were maintained just below the fusion temperature of the ash. This, however, implies that the fly-ash particles reaching the tube-sheet are essentially at the same temperature as the surrounding gas. This may be true of the very fine particles, but generally the temperature of the ash will depend on such factors as the size, velocity and residence time of the particles in the mixing section. In the hot combustion zone all the ash particles are initially in a molten state. As they pass through the mixing section, they are cooled by the dilution air, but their rate of cooling is not as rapid as that of the surrounding gas.

Also, even though the average velocity may be low enough to allow a sufficient time for cooling, local regions of high velocity exist in which the residence time of the ash is too short to allow for effective cooling. Thus, even though the gas temperature may be of the order of 1,650°F—i.e. well below the ash-fusion point—deposition is still possible and has occurred. This is the second type of deposit referred to on page 54. It occurred at relatively low temperatures (well below the initial softening temperature of the ash) and was invariably due to poor combustion in the furnace. Factors such as bad distribution of coal in the cyclone, improper fuel-air ratios and operation at too cool a temperature would result in the formation of a type of clinker. Relatively large lumps of this partly burnt-out clinker, which contains a large proportion of combustibles, would break away from the walls of the furnace and be deposited on the tube-sheet. Because of their large size they would not attain equilibrium with the gas stream and would be considerably hotter, thus being sticky and causing blockage in spite of relatively low gas temperatures.

Thus the problem of keeping the heater inlet clean is twofold. It involves:

(*a*) maintaining good combustion conditions in the furnace to prevent the formation of clinker

(*b*) keeping the heater-inlet temperature below the ash-fusion temperature of the coal burned.

Condition (*b*) imposed an operational temperature limit, which was observed in all runs after July 1954, while much of the testing time was devoted to the solution of the combustion problem. So as not to operate the plant with the tube plate excessively plugged, a resistance factor was derived which correlated such variables as the temperature, gas mass-flow and pressure drop with the degree of blockage of the tube plate. This resistance factor was found to be constant for any degree of blockage and was defined as follows:

$$\text{Resistance factor} = \frac{\Delta P}{M^2 T} \times 10^6$$

where ΔP = pressure drop of gas through exchanger in inches of water
M = gas mass-flow (lb/sec)
T = hot-exchanger gas-inlet temperature in degrees Kelvin (°C absolute).

In the clean condition, with new tube inserts, this factor was about 150. As shown in Figure 31, gradual deposition of slag on the tube plate would cause a steady increase. Usually the operation of the plant could be continued without detrimental results until the resistance factor reached 300. Values of over 400 were measured when the tube plate was severely plugged. In this condition about half of the inserts were partially or completely plugged. When they were removed, however, the tube plate underneath was invariably in perfect condition. A full new set of inserts was put in every time the resistance factor exceeded 300. Another safety measure that proved very useful in reducing the rate of deposition on the tube plate was the use of two stainless-steel screens in front of the exchanger. These were arranged as two half-circle baffles and collected a considerable amount of slag. Even though they usually became partially blocked, the increase in pressure loss was negligible (owing to the staggered arrangement). These screens had to be changed after each run, but the additional protection they offered during periods of poor combustion was well worth the slight expense involved.

4. Corrosion

The poor combustion conditions and high temperatures encountered in the tests during the first eight months of 1954 resulted in a gradual corrosion of the hot-heat-exchanger tubes.

The first indication of this corrosion was a deposit on the inside of the tubes found after 170 hours of testing. Up to that time the tubes had been clean and any particles of slag or ash could easily be removed. This time, however, attempts to remove the deposit met with little success because of the extreme hardness and strong adhesion to the tube. When first noticed, the deposit formed an even coating around the tubes and extended for about 12 inches into them. After each run, this deposit was found to extend farther into the tube. It was not until a further 100 hours had been run that the tubes were found to be leaking and it was realized that the deposit was nickel sulphide, a product of the corrosive attack of sulphur on the nickel contained in the tube metal. Areas thus attacked were gradually corroded through completely. The serious nature of the corrosion was not discovered until after 250 hours of operation. So that testing might continue, the leaking tubes were plugged at both tube plates.

Pressure tests were carried out after each run and additional leaking tubes plugged as discovered.

On the final test before the plant was dismantled for rebuilding (after 314 hours of operation) it was run with 71 tubes (14%) plugged and an additional 73 tubes (14%) leaking. In this condition the plant was still capable of delivering power.

A complete analysis of the corrosion problem is included in the analysis of results of the second series of tests. It appears on page 94.

5. Cold heat exchanger

As expected, the cold heat exchanger operated satisfactorily and, except for a seizure of the expansion joint on the first test (which was rectified by the manufacturer), no troubles were encountered. Thermal ratios of up to 0.63 were obtained in many tests (against a design value of 0.60). The pressure drop on the air side was again higher than predicted but not by such a wide margin as in the hot exchanger. There was no evidence of corrosion of any type at the conclusion of the series, and the carbon-steel tubes were found to be in good condition although maximum tube temperatures exceeded 700°F during operation.

6. Furnace

The wet-slagging cyclone furnace caused most of the troubles encountered during the first series of tests, and completely satisfactory operation was not obtained during that period. However, enough information was gained during the 314 hours of testing to make it possible to evolve a much more satisfactory design.

The cooling-water losses and their effect on efficiency have been discussed on page 48. Further reduction in combustion efficiency was caused by a high carryover of ash containing combustible materials. This in turn depended directly on combustion conditions within the cyclone. Paradoxically, highest heating efficiencies were obtained when combustion in the cyclone zone was not very good and the flame extended into the mixing zone. Under these conditions the cyclone ran cooler and the cooling-water loss was reduced sufficiently to more than offset the increase in carryover losses. However, the high carryover and the delayed combustion were detrimental to the heat exchanger and could not be tolerated for long periods of time. The lower temperatures in the cyclone also resulted in slag-draining difficulties.

Broadly, the difficulties encountered with the furnace could be classified into three categories on the basis of the functions affected, which were:

(*a*) coal feed

(*b*) coal entry

(*c*) slag draining and flushing.

These difficulties caused most of the test interruptions and are discussed in the foregoing order.

(*a*) *Coal feed*—Difficulties with the coal feed were really external to the furnace, but they had a serious effect on combustion and are therefore included here.

In the pneumatic injection system used for most of the plant operation, coal was forced in against the furnace pressure by means of a stream of air at approximately compressor-delivery pressure. At first a vibrating feeder was used to meter the coal, and the injector was only required to overcome the furnace back pressure. However, this particular feeder was not found satisfactory, and rig tests showed that the pneumatic injector could combine both operations at least as well as the dual system by control of the forcing air pressure. The feeder was discarded and both operations were performed by the pneumatic injector for all running—from 30 to 290 hours. This system was chosen because of its simplicity and its expected suitability for the plant. In Figure 34 an injector with four parallel barrels is shown.

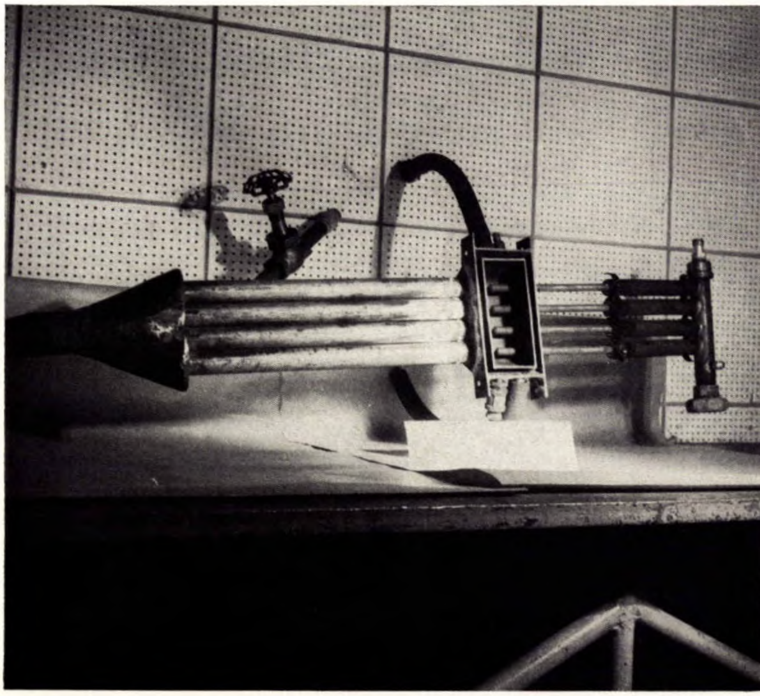


Figure 34.
Four-barrel coal injector.

Preliminary small-scale rig tests gave promising results, and some good test runs were made. The results, however, were never completely satisfactory, and operation of the plant was usually adversely affected by unsteadiness in the feed rate. The power at which the plant could be operated was often limited by the maximum rate of operation of the feed system. After about 150 hours' operation the coal-storage hopper was arranged for pressurizing to reduce the pressure difference across the injector. Steadiness was not improved, however, and it became important not to let the hopper pressure exceed the furnace pressure, particularly during transient conditions. If this happened, excess quantities of coal would flow into the furnace. Rig tests and a few test runs of the plant showed that the system was capable of operating well under ideal conditions with dry, loose coal. Experience obtained during the rest of the time showed that the system was also susceptible to variations in moisture content and the degree of aeration of the coal and that it was too sensitive to back pressure.

Several attempts were made between engine tests to carry out full-scale rig tests of the system, but no really satisfactory method of simulating engine operating conditions or of assessing the steadiness of the feed rate from the rig tests was found.

As mentioned on page 57, considerations of tube-plate blockage necessitated a reduction in the heat-exhanger-inlet temperature. This was obtained in practice by reducing the bypass flow and increasing the amount of furnace dilution air correspondingly, thus increasing the gas flow through the heat exchanger. This, of course, resulted in an increase in the furnace pressure. Other factors, such as the addition of a throat in the furnace, the plugging of the heater tubes and the fouling of the heater, contributed further to the furnace-pressure rise. As a result, the coal-injection system, always sensitive to the furnace pressure, against which the coal had to be injected, became less and less reliable.

It was therefore decided to abandon the pneumatic system and to develop a screw feeder, and for the last 20 hours the plant was operated on this system. During this short period it was not possible to evolve a completely satisfactory system of furnace entry to go with it, but some very steady operation was achieved and a modified version was adopted for the rebuilt plant.

(*b*) *Coal entry*—The development of a satisfactory coal entry proved to be one of the most difficult tasks. It was largely a process of trial and error, the 'errors' resulting invariably in poor combustion and accelerated blocking of the exchanger.

Such factors as injection velocity, direction, and proportion of primary (forcing) air had a pronounced effect on combustion.

The smaller the quantity of air injected with or around the coal, the more intense was the combustion at the coal entry. This meant that sufficient air for combustion of the coal fines reached the region of the coal entry from the main combustion air port and that additional quantities of air only chilled and diluted the combustion of these fines and volatiles. Proximity of the flame front to the entry gave good combustion with little carryover. Establishment of the flame front too close to the entry, however, resulted in an increased tendency for coke or partly burned carbon to obstruct the coal-entry pipe by forming cinderlike lumps and deposits which at times blocked the entry almost completely.

For this reason it was best to keep the flame front away from the entry, and this could be done if the proportion of primary air was increased.

The tendency towards cinder formation was also affected by the manner in which the coal was distributed on the hot surfaces of the cyclone. The air distribution was by no means uniform, and the main problem was to distribute the coal in the correct proportions. Any local excess of coal would result in the formation of cinders which would grow to a size of 2 or 3 inches and break away, sometimes covering the slag hole to a depth of several inches or being blown downstream and blocking the heat-exchanger tube plate. Many different types of entries, all injecting through a bolted flange on the closed end of the cyclone, were tried. They included two arrangements similar to those developed by Babcock and Wilcox, in which the coal is mixed with a quantity of air approximately equal to it in weight. The energy of the air is used to impart a swirl to the coal-air mixture as it enters the cyclone. The result of the first test with this arrangement was that coal coked in the entry, blocked the pipe completely and backed into the turbine exhaust-duct, where it ignited and overheated the ducting (*see* page 33).

To insure that this could not recur, the coal-injection system was separated from the turbine-exhaust air almost until the end of the test series, when a somewhat similar system was again tried. This involved the use of a smaller throat to impart a high axial velocity which would prevent the flame from flashing back into the swirl chamber, which was water-cooled by way of added precaution.

Best results were obtained with a single 2-inch-diameter pipe inclined at 35 degrees to the axis of the cyclone and directed to a point just below the main combustion air port. Rotation of the pipe only a few degrees around the center line of the cyclone would cause cinders to form, and with further rotation their formation became so severe as to prevent operation. Following success with the single inclined pipe, a similar arrangement of four pipes discharging at equal angles equally spaced around the axis was tried, but in a very short time the cyclone was filled with coke. Attempts to obtain a workable distribution by altering the direction of injection of the four pipes failed completely, and this system was abandoned. A crescent-shaped coal entry had results similar to those of the single 2-inch pipe.

The satisfactory results of the single inclined-pipe entry may be explained by the completely unsymmetrical flow pattern of the air in the cyclone, to which a symmetrical coal distribution is not suited. The air-flow pattern often seen in the frozen slag of the cyclone, and also traced by carbon deposits after operation on oil, indicated that the main path of the air within the cyclone made only three quarters of a revolution before passing through the throat. Also, a back flow under the secondary port was indicated. The best direction for coal discharge is into this back flow, by which it is distributed over the rest of the surface of the cyclone. With a symmetrical discharge of coal as given by the B & W type entries and the other arrangement, it does not seem possible for the combustion at the end of the cyclone to be as hot as that obtained with the unsymmetrical discharge. It was observed that carryover was at a minimum when combustion took place against the end of the cyclone, with fluid slag on all surfaces. As already mentioned on page 39, a temporary re-entrant throat was installed during the later runs. The results obtained when the throat was in good condition suggested that the problems of distribution could be solved and that good combustion over a wider range of flows and discharge angles could be obtained.

(*c*) *Slag draining and flushing*—Failure of the slag-removal system stopped testing more than any other cause, 17 interruptions being experienced while more than 30 different designs of the slag drain were being tested. Some of these failures were due to faulty combustion, which buried the drain hole in several inches of cinders, thus causing conditions under which no drain could be expected to operate satisfactorily. Most of the failures, however, were caused by slag freezing in the drain hole.

The drain was located in the hot zone of the mixing section in front of the first baffle because it was thought that if it were in this position, slag deposited on all surfaces would flow into it. The unsatisfactory performance of the drain hole in this position can be attributed to the following reasons:

(i) The region of the drain was evidently cooler than other parts of the furnace and did not attain slag-fluid temperatures in time to prevent the slag from solidifying as it reached the drain hole.

(ii) The slag then froze in and over the drain, thus partially insulating it from further heating.

(iii) Once frozen over, the drain could not melt out or be unblocked without outside assistance.

Various forms of assistance were tried. Mechanical means, such as ramming bars, were sometimes successful, but they seriously interfered with subsequent removal of slag from the slag box and formed an obstruction on which icicle formations accumulated. Thermal means, such as heating of the drain-hole pipe by propane combustion or electric current, were also tried at various times without much success. It was difficult to predict the fusion temperature of the slag, since the refractory materials of the furnace were slowly fluxed, the resulting mixture of refractory and slag having a melting temperature intermediate between the two constituents. On many occasions fusion-temperature checks on the substance which had frozen in the slag drain gave fusion temperatures several hundred degrees higher than the fusion temperature of the coal slag, (2,200°F). This showed that lumps of the mixture had entered the drain.

It was found necessary to drop the slag directly from a lip at or near the level of the internal surface of the refractories straight into the water. If the slag was allowed to trickle down the walls of the drain, it cooled slowly and formed icicles or failed to granulate on being quenched in water. Drains of many designs initially operated well, but when they had been in use a few hours, icicles accumulated until the drains were completely blocked. This was due to fluxing or melting of the material of the drain until its shape was changed. The result was a gradual build-up of slag on the eroded walls of the drain. Of all the refractory materials tried, silicon carbide withstood the washing action of the slag longest, but even this material slowly vanished under operating conditions. A water-cooled carbon-steel design gave the best results, but operation was always delicate because a layer of slag once frozen on top of the drain hole could not be melted out and might necessitate stoppage of the plant.

The slag flowed into a water tank below the slag drain hole and, provided it dropped into the water in a fluid state, formed granules up to $\frac{1}{4}$ inch in diameter.

By means of a hydraulic jet system, this granulated slag was flushed out through a water leg which provided a seal against the internal pressure of the slag box. As long as the slag drain operated properly, there was no difficulty in periodically flushing out the finely granulated slag. A screen was installed in the slag box to prevent the larger pieces from mixing with the small pellets and possibly blocking the drain pipe. A large accumulation of icicles and lumps, however, would block the screen and it would be necessary to stop the test in order to clean it. It was this type of trouble that necessitated the two stops shown in Figure 31.

7. Operating experience of refractories

As already mentioned, the plastic chrome-ore refractory lining in the cyclone zone was gradually fluxed away by the slag so that after a period of running the lining consisted mostly of solid slag and operated very satisfactorily as such.

The products of combustion were discharged from the cyclone at temperatures frequently in excess of 3,000°F, and the inside surfaces of the hot zone reached this figure as measured by an optical pyrometer. No material was found that would completely withstand the conditions of high temperature and slag attack, though silicon carbide was better than the other materials used in the mixing zone. Some spalling usually occurred where the penetration of the slag into the brick reached its maximum. No difficulty was experienced with refractories wherever the slag was frozen. The method of cooling the refractories by suitably routing the air entering the furnace was highly successful, and a temperature gradient of from 3,000° to 1,200°F was maintained across 4½ inches of brick without the penalty of a high heat loss.

• Conclusions

Although operation was not completely satisfactory during the first series of tests, enough information was obtained to determine the causes of the various difficulties and point the way towards their solution. Lessons learned from operating experience led to considerable modifications when the equipment was rebuilt. These are discussed in the paragraphs which follow.

1. Heat exchanger

The first series of tests clearly showed that the dry fouling of the heater surface is not as severe as was feared, thus removing the main objection to the air-tube heater. The use of this heater would allow a considerable saving in cost and size as well as the elimination of the cooled hot-end tube-sheet with its large heat loss. Unfortunately, considerations of time and cost made the construction of an air-tube heater impractical. It was therefore decided to retain the gas-tube arrangement and incorporate a number of modifications into the design.

When the heater was originally designed, high temperatures at the heat-exchanger inlet necessitated the use of a parallel-flow section to reduce the maximum tube temperatures. The experience obtained during the first series of tests showed that lower gas temperatures than originally planned had to be used in order to prevent fouling of the exchanger inlet. The resulting decrease in maximum tube temperatures made the parallel-flow section (with its accompanying high pressure loss) unnecessary. Thus a straight counterflow exchanger could be used, and this gave promise of a considerable reduction in the air-side pressure drop.

The limit imposed on the heat-exchanger-inlet temperature required a sizable increase in the furnace dilution air, and this would reduce the bypass stream considerably. The next logical step was to eliminate this bypass stream and pass all the gas through the heat exchanger. In order to effect a further reduction of the pressure losses in the system it was decided to eliminate the cold heat exchanger even though this would reduce the total heating surface and consequently the thermal efficiency (25). To obtain more information on corrosion, resistance tubes of various materials were to be installed in the new exchanger.

2. Furnace

The main conclusions obtained from the results of furnace operation were the following:

(a) To insure good combustion, a re-entrant throat would have to be used in spite of the high pressure losses associated with it.

(*b*) The cooling-water loss could be reduced some 30 per cent by eliminating the mixing-zone baffles. This would also offset to some extent the additional pressure loss caused by the re-entrant throat. In view of the protection provided by the ceramic inserts, there would be no need to fear direct radiation from the combustion zone to the heat exchanger.

(*c*) The slag drain hole should be placed in the cyclone section to insure that it would operate sufficiently hot to remain open. With this arrangement some slag could spill over the throat and into the mixing zone, but it was expected that the increased volume of this zone would permit deposition of some slag without deterioration of performance.

(*d*) The mixing section would be made as large as possible and the dilution air introduced as close to the cyclone throat as possible. Thus oxygen would be available to complete, at the earliest moment, the combustion of any combustible gases being discharged from the cyclone. The large volume and cross-sectional area of the mixing zone and the symmetrical gas flow would insure a uniformly low gas velocity and the lowest quantity of airborne slag particles while increasing their residence time in the mixing zone.

The early injection of the dilution air would exclude from contact with the hottest gases all but water-cooled surfaces. The refractory bricks would thus operate at lower temperatures with prospects of longer life. It would also be possible to use cheaper refractory materials.

PART **3**

SECOND SERIES OF TESTS
(JANUARY-DECEMBER 1956)

DESCRIPTION OF REBUILT PLANT

• General Considerations

The conclusions outlined in the previous chapter served as a basis for the modifications which the plant underwent before the second series of tests. The general arrangement of equipment was retained, with the power unit on the mezzanine floor and the combustion and heat-transfer equipment underneath (Figure 35 a, b and c). The air intake was not changed, but the compressor-outlet manifold was rotated so that it discharged the air upwards into a rectangular 90-degree bend and transition section connecting to a 16-inch-diameter transfer duct which delivered the air to the cold end of the heat exchanger (Figure 35c). A flexible joint was installed in the transfer duct to prevent movement of the manifold due to any expansion of the heater.

The exhaust gas leaving the air heater was turned through 180 degrees and ducted to the exhaust stack. A water-filled fly-ash trap was installed just ahead of the exhaust stack to collect any fly-ash remaining in the gas stream.

• Air-heater

As shown in Figure 36, the heat exchanger was rebuilt without the parallel-flow section. The old carbon-steel shell was used, but the hottest section, which corresponds to the previous parallel-flow section, was made of stainless steel, and the 3-nozzle air-outlet ring was placed directly behind the tube-sheet. The tube-sheet water-cooling had proved so effective during the first series of tests that both the main and the auxiliary tube-sheets were now made of carbon steel. The rear surface of the main tube plate was again air-cooled. Air for this purpose was bled from the compressor delivery pipe, passed through a water-cooler and a booster fan, and admitted into the heater through two ports located directly behind the tube plate. The shell of the heat exchanger was lined on the inside with a 2-inch thickness of asbestos insulation in order to reduce the heat losses and to keep the metal temperature at a safe value.

A considerable amount of time was spent on the design of the tube bundle. A small-scale model of a section of the bundle was made and various baffle arrangements were investigated to obtain the configuration that would yield a uniform flow distribution and a minimum of pressure loss. As a result of this investigation, disk and doughnut baffles were retained and the tube holes were drilled $\frac{1}{16}$ inch oversize to reduce as much as possible the high pressure loss produced by cross-flow within the bundle. Extra space was provided between the tubes and the inner duct in the region of the outlet nozzle in an effort to insure uniform flow near the tube-sheet.

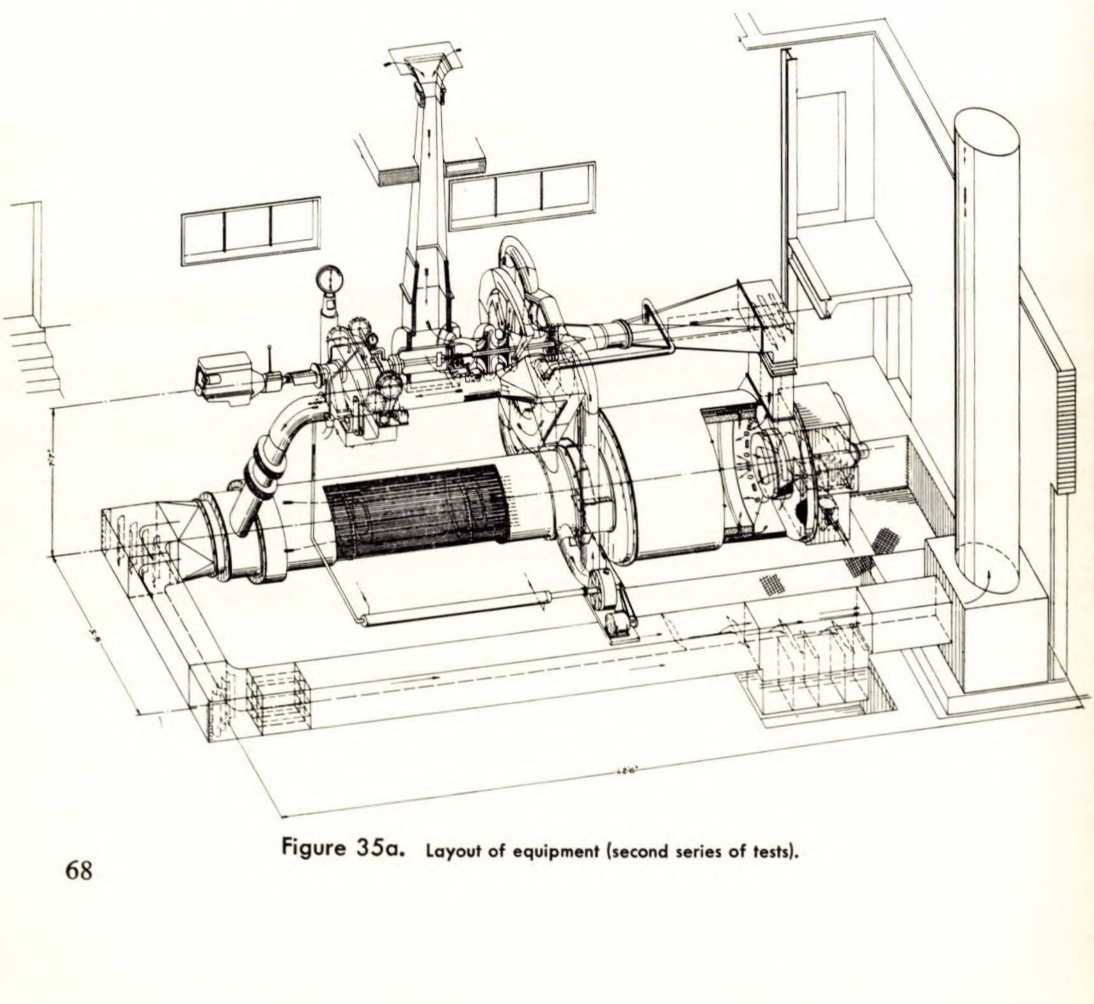


Figure 35a. Layout of equipment (second series of tests).

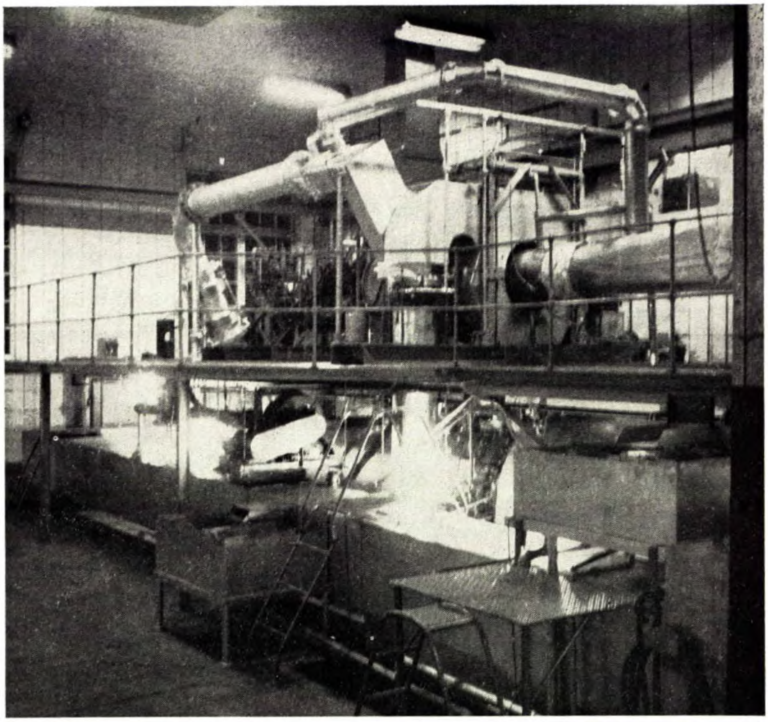


Figure 35b.
Rebuilt plant, general view.

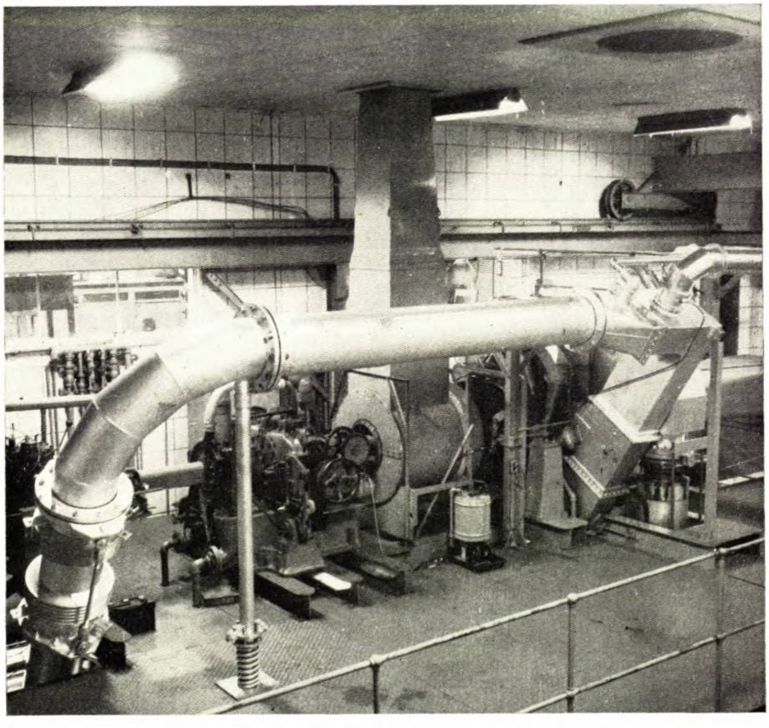


Figure 35c.
Rebuilt plant, machinery line.

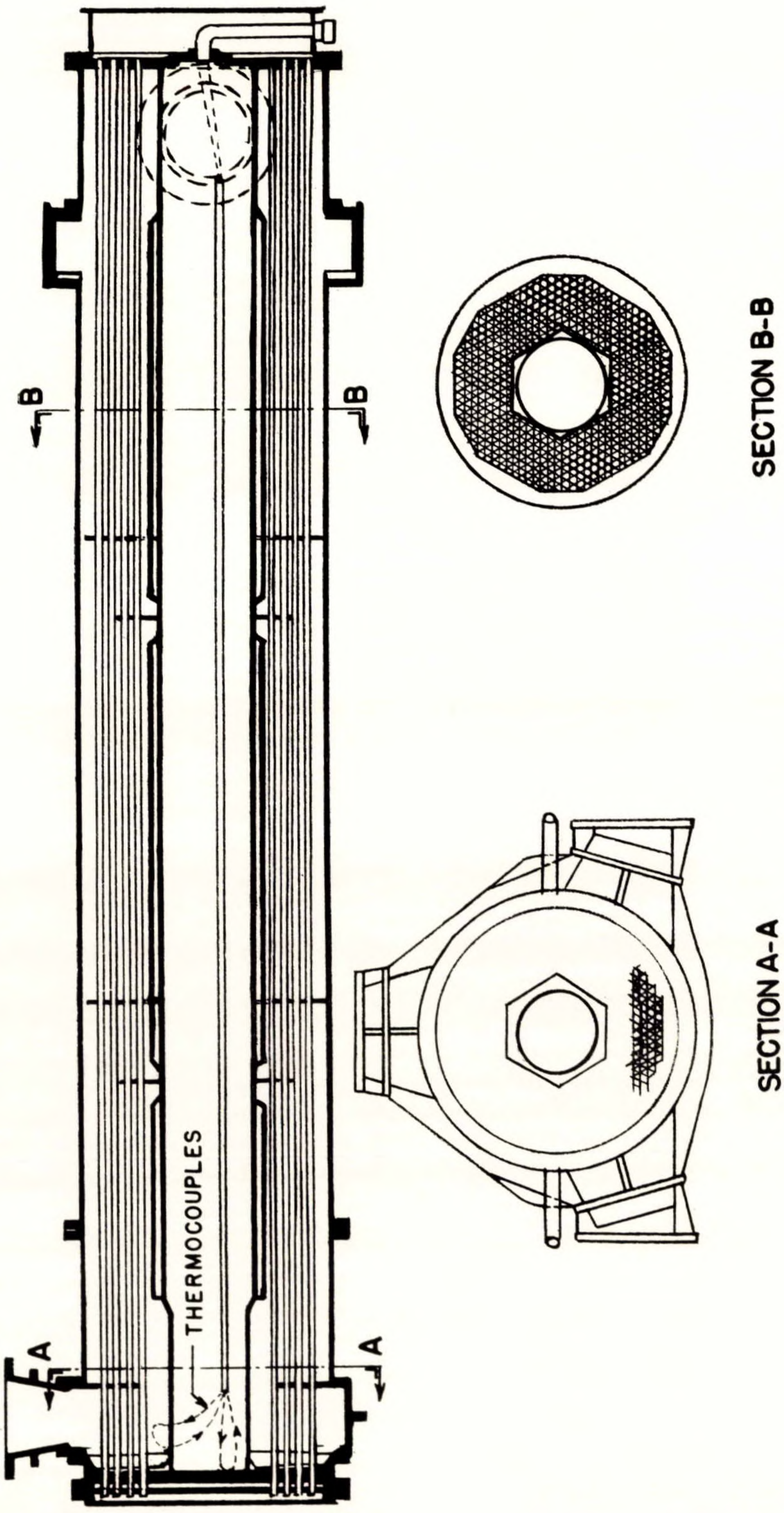


Figure 36. Air-heater (second series of tests).

The tube-length remained the same as in the original heater. In order to reduce the cost of retubing, use was made of the Nimonic tubes from the original heater. Since these tubes showed evidence of corrosion near the hot tube-sheet only, they were cut off 7 feet from the hot end and butt-welded to 7-foot lengths of carbon-steel tubes. The carbon-steel section was then placed at the cold end of the heater. Thermocouples were attached to the tubes at the hot end and near the welded joint. The number of tubes was increased from 498 to 500. In order to investigate resistance to corrosion of various tube materials, sets of tubes of 12 different alloys were interspersed throughout the bundle.[32] It was expected that trouble would occur owing to the differences in the expansion rates, but this was deemed a small penalty to pay for the information on resistance to corrosion that would be gained. The heater was again fully instrumented so that all vital temperatures and pressures could be measured. The instrument leads were taken out of the shell through the central duct and rear tube-sheet or through the air nozzles.

• Furnace

The new design of the furnace differed considerably from the original (Figure 37). The cyclone section was the one used previously, but it was installed horizontally and featured a detachable 12-inch-diameter water-cooled re-entrant throat at the exit from the section. The slag drain was placed in the cyclone section. Originally it was at the mid point of the cyclone drum on the center line, but later it was moved closer to the throat and slightly off center in order further to improve slag-draining. Water-cooling was omitted from the section of the cyclone wall surrounding the slag drain in an effort to maintain high temperatures so that the slag in this region would melt first and freeze last during transient conditions.

A large flange section was added to the throat of the cyclone. It was bolted to the upstream end of the mixing zone and formed an air plenum chamber surrounding the furnace (Figures 38 and 39). The turbine-exhaust air which entered this plenum chamber was split into combustion air, admitted tangentially to the upstream end of the cyclone drum, and dilution air, which entered the mixing section immediately downstream from the cyclone throat through a series of openings in the conical wall separating the plenum from the mixing zone. These openings had mechanically operated shutters to give some control of the dilution air. Ports were also provided in the plenum chamber to admit air to the brick cooling passages in the wall of the mixing section.

The inlet of the combustion-air passage was in the shape of a rectangular bell-mouth nozzle (Figure 38). The passage was fitted with a damper control and, to make possible the accurate calculation of the air-fuel ratio during operation, the intake was carefully calibrated.

[32] Their location is shown in Appendix 9.

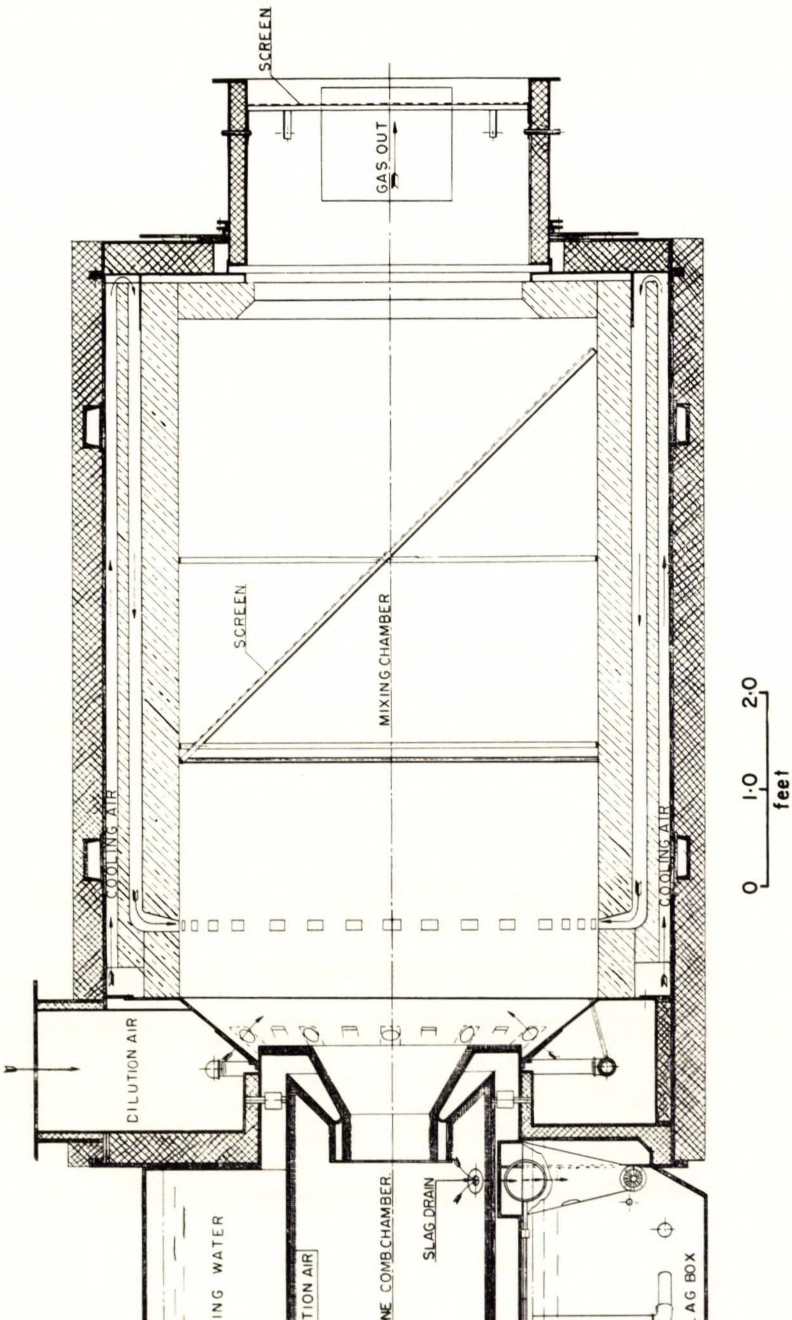


Figure 37. Furnace (second series of tests).

The cyclone drum was made easily detachable from the mixing section and was suspended on overhead tracks so that it could be rolled back for easy maintenance.

The mixing section consisted of a cylindrical mild-steel casing lined on the inside with 4½ inches of insulating brick and another 4¼ inches of fire-brick. Passages for cooling air were provided in the refractory materials, the use of a carbon-steel casing being thus permitted. The slightly higher pressure of the cooling air insured that any leakage that might develop would be that of cool air into the furnace rather than of hot gases outwards.

Complete omission of water-cooled baffles reduced the cooling-water losses by about 30 per cent. The inside diameter of the mixing section was 4 feet 2 inches, and the length 7½ feet.

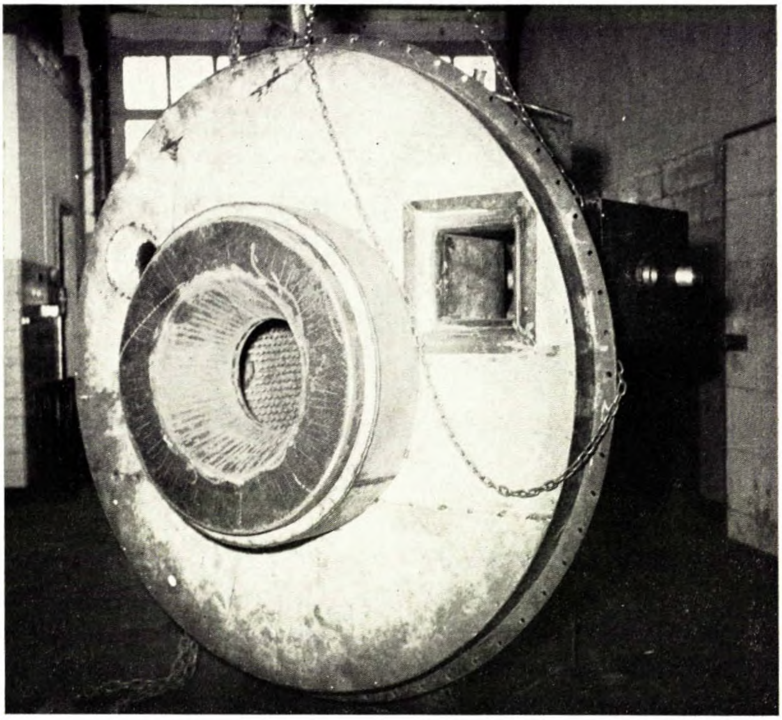


Figure 38. Detachable cyclone assembly.

In order to reduce slag deposition on the air-heater tube plate, stainless steel screens were placed across the mixing section.

Access to the heater inlet and the mixing section was provided by two doors placed in the connecting duct. One of these doors also served as the mounting for a gas-sampling probe and its electrically driven traversing gear.

• Coal-handling Equipment

The coal-handling equipment was essentially the same as in the first series of tests. A magnetic separator was installed in the coal elevator to remove metal particles from the coal, and a coal-drying system operating on turbine-exhaust air was incorporated in the processing circuit. This was made necessary by the deleterious effect of excessive moisture on the free movement of coal inside the scale hopper and consequent difficulty in maintaining a constant feed rate into the cyclone.

To improve the free flow of coal from the hopper, the inside slope of the walls was increased by the use of aluminum sheeting, and a vibrator was mounted on the hopper wall.

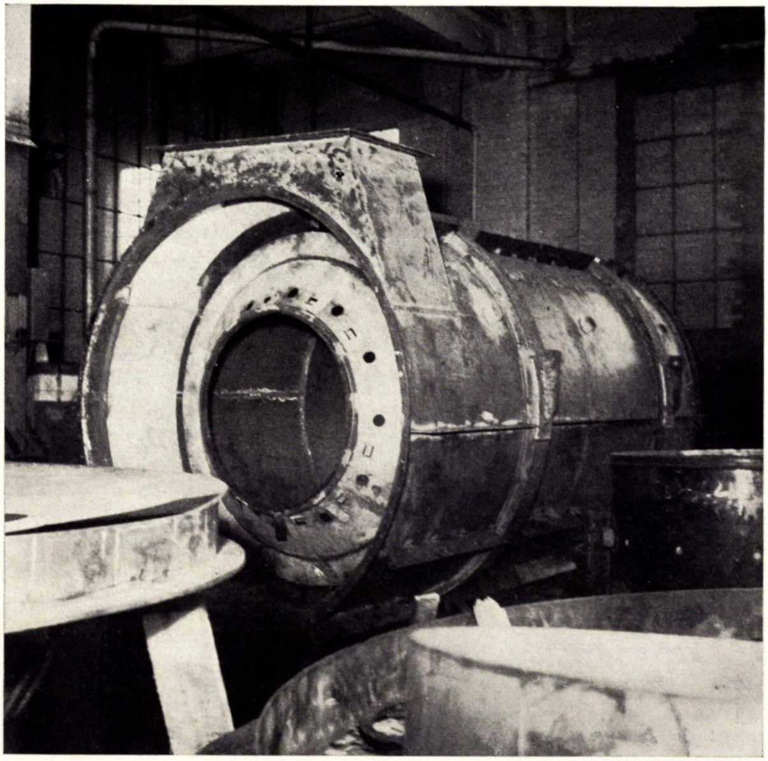


Figure 39. Mixing section during assembly.

Towards the end of the first test series a feed-screw method of metering and feeding the coal showed promising results. It was extensively experimented with during the rebuild of the plant and, although actual firing conditions could not be duplicated exactly, enough information was obtained to design what promised to be a successful system. It is shown in Figure 40. To the bottom of the scale hopper was attached a box section, from which a 4-inch feed-screw delivered

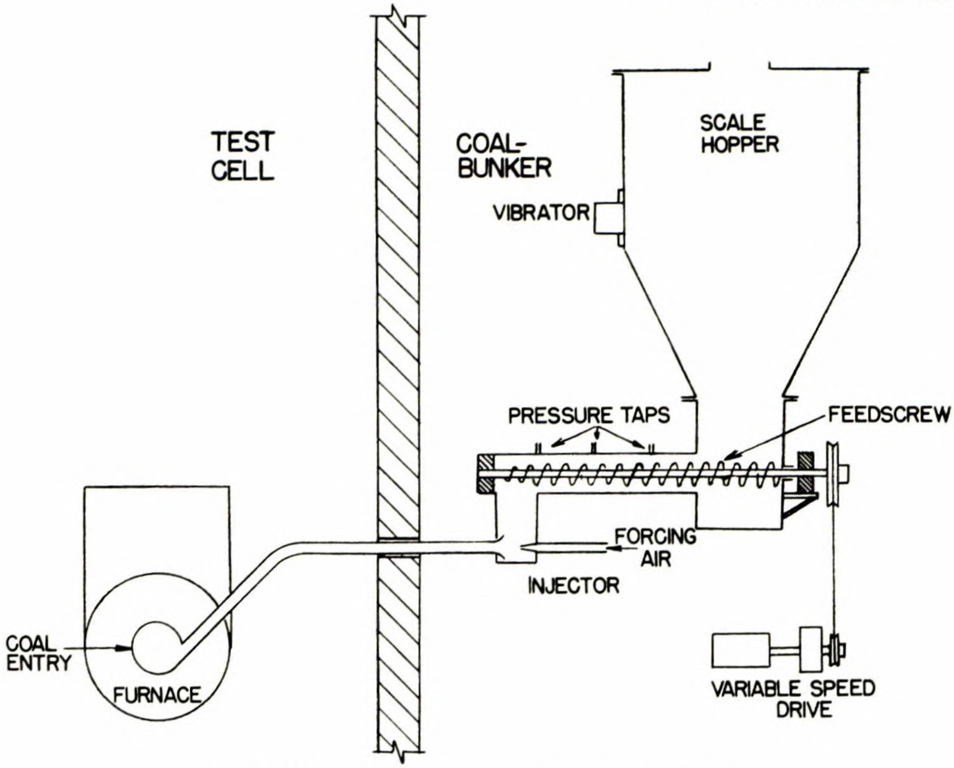


Figure 40. Coal-feed arrangement (second series of tests).

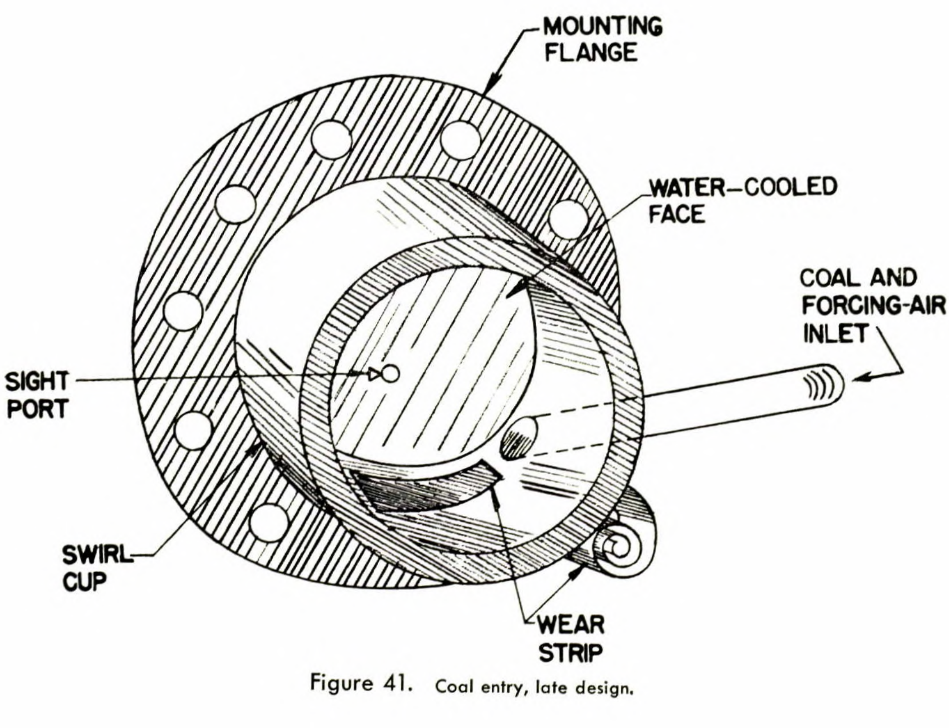


Figure 41. Coal entry, late design.

the coal to the injection box. Compressed air from the gas-turbine set was used to convey the coal particles to the furnace. Metering was obtained by varying the speed of the feed-screw.

Various refinements, such as injection of fluidizing air into the screw-pipe (to prevent jamming of the feed-screw) and the addition of inflatable hopper bags and air jets inside the hopper (to prevent arching of the coal and consequent starving of the feed-screw), were added to this system as testing experience dictated. The development of a satisfactory coal entry had to await engine tests, and many designs were tried once testing got under way. The design which gave best results and was finally adopted consisted of a swirl cup (Figure 41), with a water-cooled face. The inside of the cup was protected from abrasion by a wide strip of spring steel inserted circumferentially and moved periodically as it became worn by the coal particles in the incoming stream.

TEST SERIES

The experimental work was resumed in January 1956 after an 8-month period of redesign and rebuilding. The rebuilding period took more time than expected owing to continuous delays in the work on the air-heater, which was subcontracted to the firm that built it originally. After a few preliminary runs on oil (to test for leaks and dry out the insulation), the plant was run on coal on January 9 and 12. The runs were short (of less than 10 hours' duration) because of poor combustion, feed-screw jamming, and difficulties with slag removal and flushing. A modification of the throat re-entrance[33] resulted in a considerable improvement in slag- draining, and on January 18 a test of 30 hours' total duration was run. It was interrupted several times so that slag drains of different kinds, including some of graphite- and silicone-tube design, could be tried, but none worked satisfactorily. Gradual corrosion of the drain material caused the slag to run down the sides of the drain-pipe (rather than drip off the lip), and under these conditions freezing and eventual blockage were inevitable. Another source of trouble associated with poor draining was that large icicles of frozen slag, knocked off the drain-pipe by a mechanical poker to prevent blockage, accumulated in the slag box and prevented proper flushing. Screens placed in the slag box for the purpose of collecting these larger pieces were soon blocked completely, and test runs had to be interrupted to remove them and clean them out.

Some 225 hours of testing elapsed before these difficulties were finally overcome and satisfactory slag removal was obtained. This was achieved by a careful study of the slag-flow pattern in the cyclone and resulted in the relocation of the slag hole. The new location was made coincident with the area of heavier slag build-up, close to the cyclone throat and slightly to the side of the center line. The slag in this area was generally quite fluid and drained well.

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[33] Figure 37 shows the furnace with the modified throat. The original design consisted of the conical throat only and was modified by adding a 6-inch-long cylindrical section extending back into the cyclone.

Although it was impossible to prevent occasional freezing and blockage of the drain hole, there was no difficulty in reopening it with a specially designed poker inserted from underneath and actuated manually from outside the slag box.

Another difficulty, which caused most test interruptions in the first few months of 1956, was the unsteadiness of the coal feed to the cyclone. This resulted from the

(*a*) unsatisfactory performance of the coal entry

(*b*) jamming of the feed-screw or a loss of feed followed by blow-back to the hopper.

Many different coal entries were again experimented with. Modified versions of the B & W type were tried with various amounts of primary air, but heavy cinder formation again caused poor distribution of coal in the furnace and resulted in excessive carryover and rapid fouling of the air-heater tube plate. Swirl type entries were next tried and after various sizes and arrangements had been investigated, the design shown in Figure 41 was finally evolved. The stream of coal particles at first subjected it to fairly rapid erosion, which caused a gradual deterioration in performance. This was overcome, however, by inserting a strip of hard steel as explained on page 76.

The rear face of the swirl cup, which was exposed to direct radiation from the flame, was water-cooled. In one of the tests [34] the cooling proved inadequate and the metal was burned through, water being thus allowed to enter the cyclone fast enough to prevent the coal from igniting properly. The operator discovered this when the furnace-outlet temperature and the engine speed decreased in spite of his efforts to increase the rate of coal feed. When the temperature and speed continued to fall, the fuel was turned off and the plant stopped. Examinations of the furnace revealed the cause of the trouble. The water entering the cyclone gradually extinguished the flame and caused the slag drain to freeze over so that the cyclone was found flooded to a depth of about 3 inches. No damage occurred, however, and the coal entry was repaired and the plant was started again in about 90 minutes.

Further experimentation with different sizes of swirl cups finally resulted in the adoption in June of the most reliable design, which permitted non-stop runs of 100 and 200 hours to be carried out successfully.

While a satisfactory coal entry was being developed, considerable effort was directed towards perfecting the reliability of the coal-feed system.

The satisfactory operation of the coal feed-screw depended on a number of factors. Until these were isolated and analyzed, the coal-feed rate remained erratic.

The first factor was the tendency of the coal particles to jam in the screw-pipe, thus increasing the torque on the screw and causing the overload relay to trip and stop the screw completely. This was overcome by carefully selecting the clearance between the screw and the pipe and by injecting air into the pipe in order

[34] Test of May 21-24. Total duration 64 hours.

to fluidize the coal stream and thus facilitate the flow. A series of pressure taps was installed on the pipe, and an indication of satisfactory operation of the screw could be obtained in the control room by observing the pressure gradient along the pipe.

The second factor, diametrically opposed to the first, was the frequent starving of the feed-screw. This was invariably caused by improper flow of coal from the hopper. Coal would either stick to the sides and leave a completely clear funnel in the center, or it would arch over the opening at the bottom of the hopper. In either case the feed-screw would run empty and the injector air and furnace gases would blow back along the screw and into the hopper.

The flow could be re-established by agitating the coal in the hopper, and such means as inflatable bags, air jets directed into the hopper and mechanical pokers were tried at various times in an effort to accomplish this automatically. The best results were finally obtained by mounting a small vibrator on the side of the hopper and lining the hopper with aluminum sheets. The increased slope of the walls and their smooth surface minimized arching or sticking of the coal.

One of the main factors preventing the free flow of coal and controlling its tendency to cake is its moisture content. It was found that best flow characteristics were obtained when the coal was from ½ to 1 per cent wet. Owing to outside storage and the variety in the types of coal used, the moisture content varied considerably from test to test.[35] In order to overcome this, the warm test-cell air augmented by small quantities of turbine-exhaust air was used to dry the coal. This warm air was admitted to the riser pipe, which conveyed the coal to the rotoclone. When drying was not required, cold air could be introduced into the rotoclone circuit by a simple damper adjustment.

The improvements to the coal-feed system gradually resulted in a marked improvement in combustion and a much steadier operation. This was evidenced by a steady lengthening of test duration times and a reduction in the frequency of stoppages. The first 50-hour non-stop run took place on April 25 and 26 and was followed, from June 21 to 26 by a 100-hour run.

By that time, 481 hours of operation had been logged (all but 12 of them on coal) and the number of heat exchanger tubes made unserviceable by corrosion gradually increased. The run of June 21 was made with 21 tubes plugged, and a further 44 Nimonic tubes were found leaking after the 100-hour test. Since this was equivalent to a loss of nearly 14 per cent in heat-transfer surface, it was decided to replace the defective tubes rather than continue to plug them. Accordingly, the heater was disconnected from the ducting and moved to another part of the test cell where the work could be carried out conveniently. Most of the defective tubes were replaced with Type 446 stainless-steel tubes, which up to that time appeared to have best withstood corrosive attack. For quick delivery, tubes with a thicker wall (16-gauge) and hence a smaller inside diameter were

[35] A considerable amount of work was done on development of an electrically operated coal-moisture indicator. This work resulted in the construction and successful operation of a capacitance-type moisture meter working at a frequency of 1,000 cycles per second (33).

accepted, so that the ceramic inserts could not be used with them. Accordingly, stainless steel inserts were made and used with these tubes.

Samples of the corroded tubes removed from the heater were again sent to the Mines Branch for analysis.

Retubing and other plant modifications were completed, and testing was resumed in October. A run of 134 hours was made from October 11 to 15. It was characterized by long periods of steady operation with coal-feed difficulties reduced to low incidence. A stop was made after some 91 hours in order to clean the stainless-steel screen installed in the mixing zone. Although the large section of the mixing zone prevented large lumps of slag from being directly airborne, it was found during the test that small slag particles conglomerated to form larger lumps on the furnace wall and that these lumps broke from the roof of the furnace and were blown towards the heater while falling.

The screen prevented them from reaching the tube plate but became partly blocked in the process, thus increasing the gas-pressure drop undesirably. So that the plant could continue to operate even with the screen blocked completely, the screen was modified after the test of October 11 to consist of two sections arranged as overlapping baffles far enough apart to give a minimum pressure drop. This arrangement has proved very successful and has not reduced the collection efficiency of the screen to any appreciable degree. As an added precaution, a second screen was installed directly in front of the tube plate to catch agglomerated ash from the thermocouples and other surfaces. Although this screen had a much smaller mesh than the main screen, it never became so obstructed as to affect operation except when a defect developed in the first screen.

The test of October 11-15 was finally stopped by a failure of the Dart compressor rotor. The failure was attributed to excessive operating time—875 hours at the Gas Dynamics Laboratory in addition to a considerable amount of previous operation. A new engine was installed (Dart No. 13), and another test was started on October 31. This proved to be the most successful test run of the whole program. It lasted more than 10 days, and 247 hours 24 minutes of it, including more than 200 hours of non-stop running, were logged on coal.

A stop was made during the test to determine the response of the plant during rapid shut-down and starting, and the results are analyzed on page 84.

Thirty-nine Nimonic tubes developed leaks during the run and had to be plugged after the test in addition to eight tubes which were found defective after the previous run.

A run of about 4½ hours was made on November 22, mainly to demonstrate the operation of the plant to delegates of the Conference on Coal-burning Gas Turbines, which convened in Montreal on November 22 and 23.

This effectively terminated the second series of tests, as further running would have required replacement of the corroded tubes, their number having risen to 82 after the previous run. Since a full analysis of the performance of the various tube alloys was also urgently required, the air-heater was disassembled so that samples for analysis might be obtained.

ANALYSIS OF RESULTS

• General Considerations

1. Running time

The second series of tests was characterized by a continued improvement in the performance and reliability of the various components of the plant. This is well illustrated in Table V, which shows a breakdown of testing time. 'Motoring' refers to the time during which the plant was motored on the starting engine for various checks and calibrations and in preparation for the lighting of the furnace at the start of a test. 'Assisted' refers to operation with the furnace lit but with the starting engine connected and supplying some power. This type of running occurred during the starting period when the plant was being gradually accelerated to self-sustaining speed. The drastic reduction in assisted time during the second series of tests is due to the greatly improved starting procedures. 'Oil' represents the time during which it was impossible to burn coal owing to various coal-feed difficulties and also the time at starting before coal operation began.

Table V

Summary of Testing Time

	First Test Series	Second Test Series			Combined
	Nov. 53 Apr. 55	Jan. 56 June 56	Oct. 56 Dec. 56	Total (Jan.-Dec.)	Total 1953-56
Motoring	244:54*	39:45	7:59	47:44	292:38
Assisted	71:32	49:54	9:03	58:57	130:29
Total	316:26	89:39	17:02	106:41	423:07
Oil	83:15	12:12	1:10	13:22	96:37
Coal	230:50	468:41	387:02	855:43	1,086:33
Total	314:05	480:53	388:12	869:05	1,183:10
Total testing time	630:31	570:32	405:14	975:46	1,606:17

*The time is given in hours and minutes.

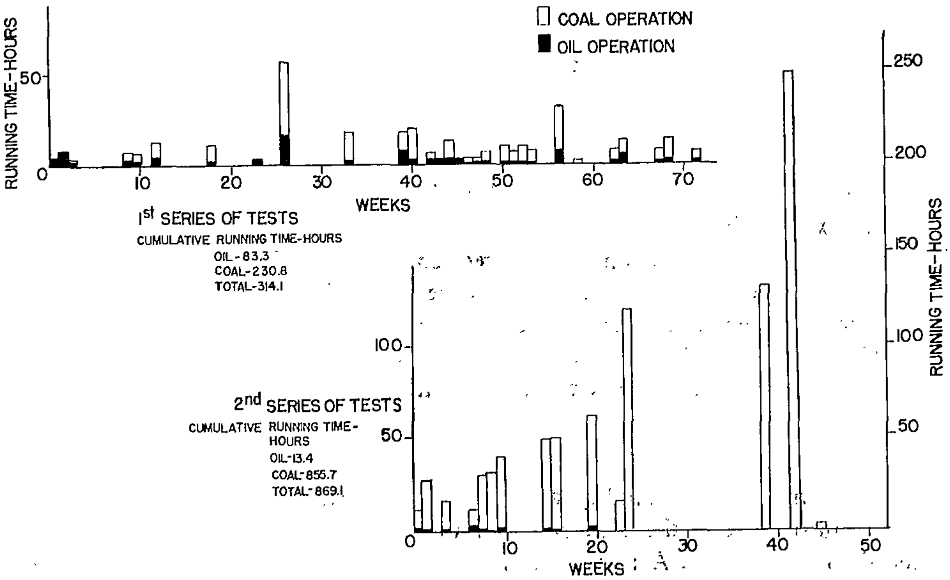

Figure 42. Plant running time.

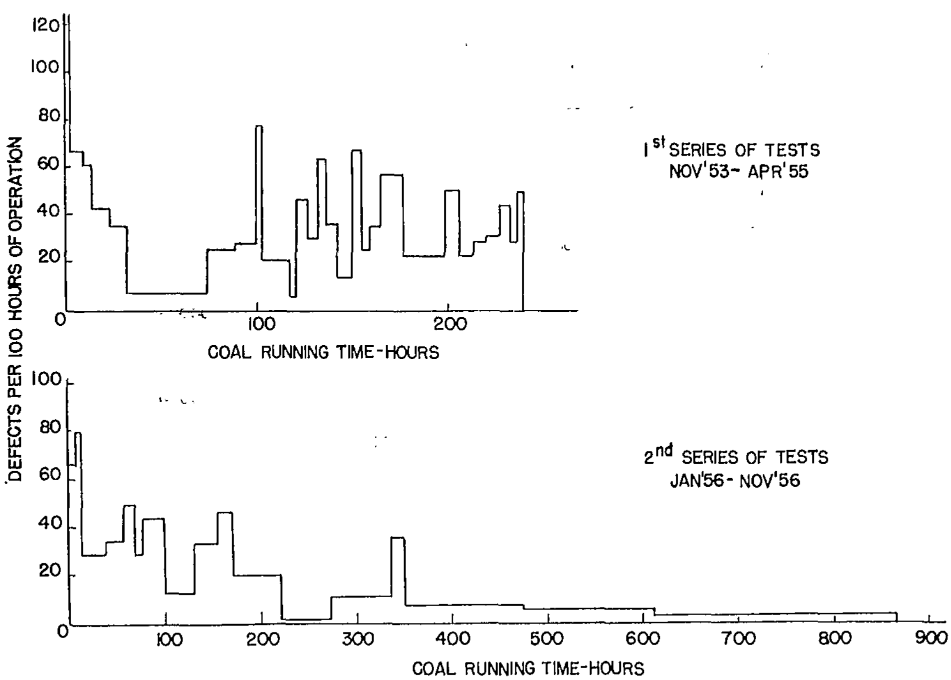

Figure 43. Plant defects.

The three longest non-stop runs lasted 200 hours 50 minutes, 102 hours and 86 hours respectively.

The coal burned during the second test series totalled 822,763 pounds at an average rate of 961 lb/hr.

A detailed breakdown of the testing time is shown in Figure 42, which clearly illustrates the greatly improved performance obtained with the rebuilt plant.

The increased reliability of the plant is also well illustrated in Figure 43, which shows the frequency of defects and difficulties encountered during testing. This refers to difficulties which required some form of corrective action but which in about 60 per cent of the cases were rectified without stopping the plant.

It can be seen that in the last 500 hours of operation the average is about 4.7 difficulties per 100 hours, which corresponds to average trouble-free periods of 21 hours. In the first series the average trouble-free period of operation was about three hours.

2. Control characteristics

The control and flexibility achieved in the first series of tests have been greatly improved, and the time required to start the plant from cold has been reduced to 40 minutes. A start of this type is shown in Figure 44. The furnace is lit and the plant is brought up to temperature on the cheapest distillate oil readily available.

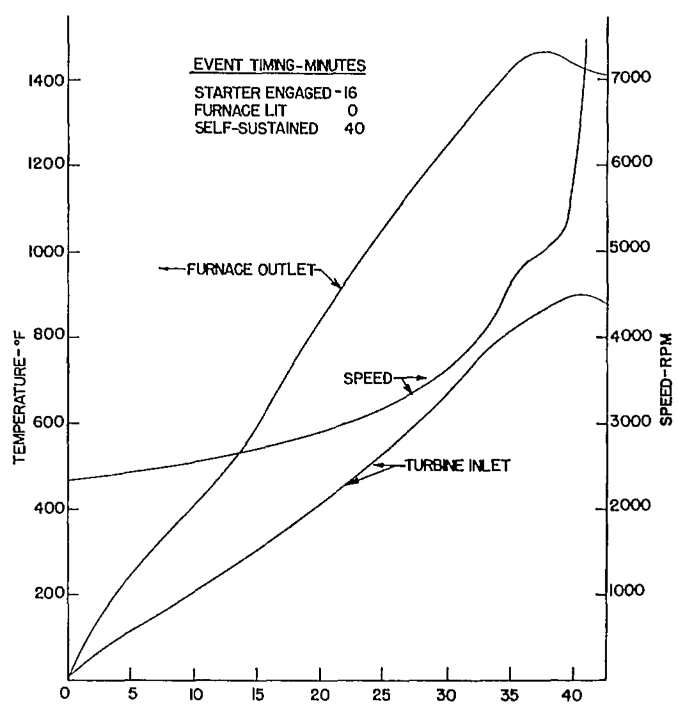

Figure 44. Transient conditions during a start from cold.

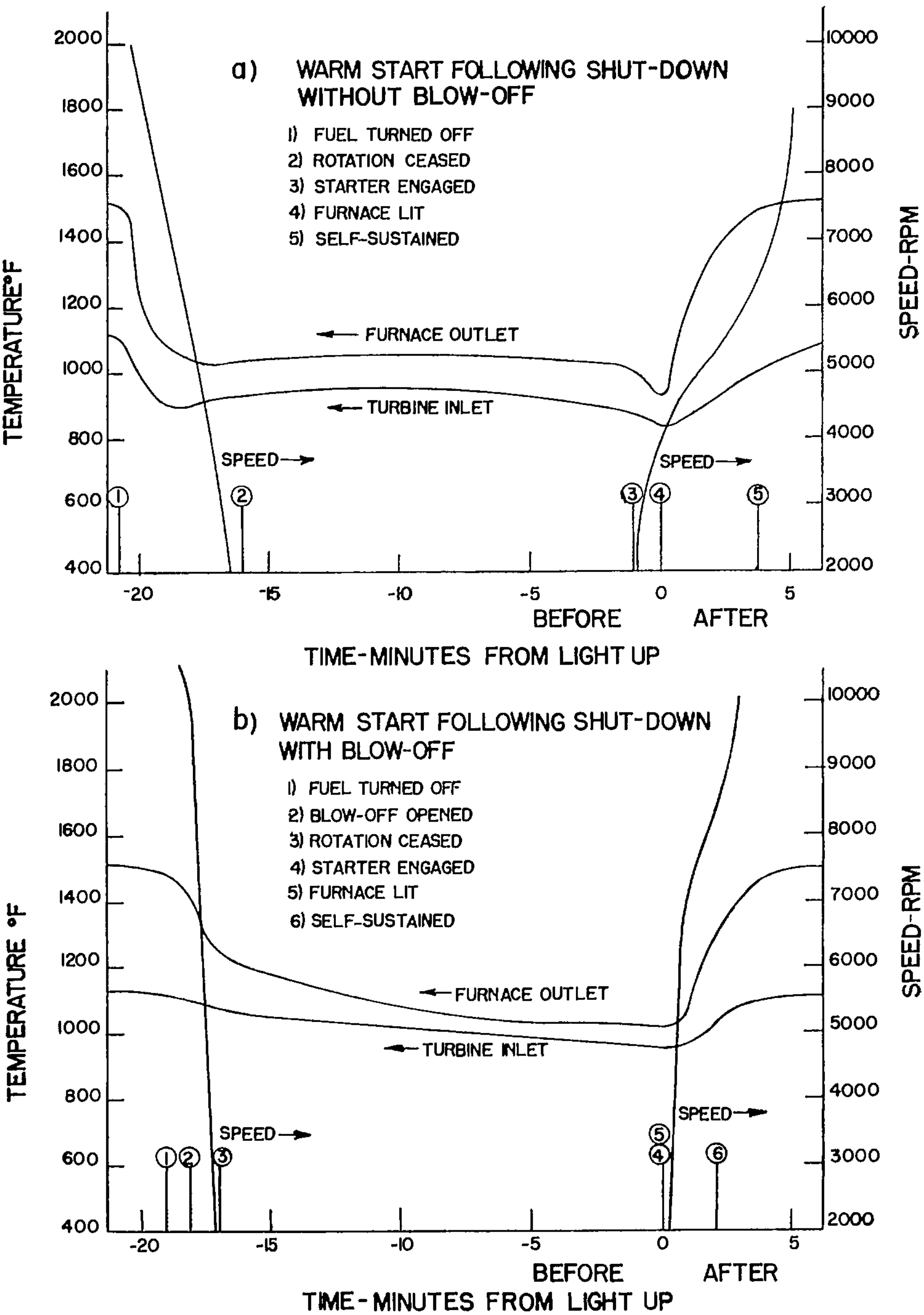

Figure 45. Transient conditions on rapid stops and starts.

After the correct throttle setting of the starting engine has been selected, no further adjustment is required. If a free wheel were incorporated, the turbine would accelerate as soon as it was ready without any manual operation, but in practice the operator declutches the starting engine when its manifold pressure indicates that it is delivering no power.

Coal is fed into the furnace during the rapid acceleration which occurs as soon as the turbine becomes self-supporting and the starting engine is declutched. Thus, practically no oil is used during self-sustained operation.[36]

When the plant is hot, the restart procedure is similar. The operator selects the starting-engine throttle setting and fuel flow which correspond to the prevailing turbine-entry temperature. After that, the starting requirements are the same as during a start from cold.

The actual temperature levels prevailing during a 'warm' start depend on the method of stopping the plant. Normally, when the fuel is turned off, the gas turbine will continue to rotate for about five minutes, circulating the air and cooling fairly quickly (Figure 45a). If, however, the blow-off valve on the compressor delivery is opened, the turbine stops in about a minute, and temperatures remain at considerably higher levels (Figure 45b).

Even after a stop of 20 minutes, the plant has been restarted in less than two minutes without any difficulty. This characteristic of the exhaust-heated turbine would become particularly important in locomotive line service, where the large fuel consumption at idling would no longer be a disadvantage, the fuel being shut off completely during stops of less than about 30 minutes' duration.

3. Thermodynamic performance of plant

A detailed analysis of the performance is presented here for a representative run. The test analyzed is that of May 21-24, 1956, during which additional observations and temperature traverses were made to allow an accurate assessment of performance.

The run was of more than 61 hours' duration, and the final condensed tabulation represents results of approximately 100 readings obtained during periods of steady operation.

The average test conditions were as follows:

Engine speed	11,520 rpm
Load	101 hp.
Coal rate	961 lb/hr
Furnace-outlet temperature	1,522°F(828°C)
Turbine-inlet temperature	1,133°F(612°C)

Table VI gives a summary of the heat balance in the plant. Details of the calculations and a more detailed breakdown of various losses are given in Appendix 10. The complete heat balance is also shown graphically in Figure 46.

[36] A comparison of this sequence with the original starting procedure described on page 33 may be of interest at this point.

Table VI

Heat Balance of the Coal-burning Plant
(basis: heat supplied in coal)

—	Btu/sec	Percentage
Heat supplied in coal	3,727	100
Heat to useful power	79	2.13
Heat to cooling water	832	22.3
Ventilation loss	369	9.9
Miscellaneous losses (lubricating oil, coal-drying)	16	0.43
Heat to exhaust	2,458	66.0
Total	3,754	100.8

Two aspects of this heat balance are particularly worthy of notice: firstly, the excellent agreement between the summation of the individually measured losses and the heat supplied in fuel; secondly, the low over-all thermal efficiency.

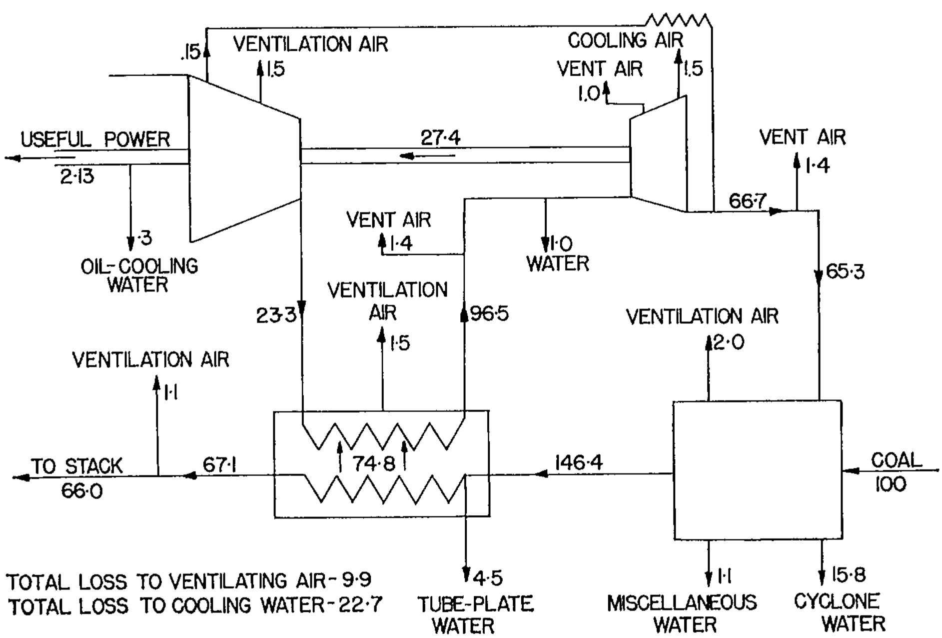

Figure 46. Typical heat-balance diagram (second series of tests).

As regards the first, the calorific value of the coal was obtained from measurements made on five samples and ranged from 13,000 to 14,070 Btu/lb (gross). This was used as a basis for the heat-input calculation.

The losses were measured individually except the ventilation loss, which represents the heat lost by convection and radiation from exposed surfaces. This was obtained by installing a powerful suction fan above the roof of the test cell. The roof opening was made into a sharp-edged orifice carefully calibrated so that the flow of ventilating air could be measured accurately. Cold air entered the test cell through openings at floor level. Measurement of the over-all temperature difference made it possible to calculate the over-all ventilation loss.

This loss was then apportioned to various components of the plant by making an estimate of the size and temperature of the various surfaces.

The ventilation loss amounts to nearly 10 per cent of the input. This is rather high, but can be attributed, firstly, to the dispersed layout of the plant and, secondly, to the vigorous circulation of ventilating air required to keep the test-cell temperature down so that such functions of the operating crew as constant visual observation of the furnace and the taking of temperature traverses can be performed without too much discomfort.

The greatest loss (66 per cent of the total) was the exhaust-stack loss. This was caused by the high gas-outlet temperature from the air-heater, which averaged 735°F.

It was, of course, realized that the removal of the cold heat exchanger would reduce the heat-transfer surface by more than 50 per cent and hence affect the efficiency of the plant. Some compensation for this would be obtained from the slight reduction in pressure drop and a slight increase in the heat-transfer coefficient in the remaining exchanger (due to a higher gas flow through it), but a reduction of efficiency was expected.

The second aspect of the heat balance that requires comment is the low over-all thermal efficiency of 2.13 per cent, which is already partly explained by the removal of the cold heat exchanger. This, together wtih the limit of 1,560°F (850C°) imposed on the furnace-outlet temperature, reduced the turbine-entry temperature to 1,130°F (612°C), thus causing reduction in the work output and hence in thermal efficiency. It is estimated that, had funds been available to rebuild the heat exchanger as a single unit with an area equal to the total original area (i.e. with two exchangers) and such a distribution of surface as to obtain the same pressure drop as with the present unit, the turbine-inlet temperature would have been raised to 1,262°F (684°C) and the gas-outlet temperature reduced to 617°F (325°C), with a resultant decrease of nearly 18 per cent[37] in the exhaust-stack loss.

Thus it can be seen that, had a high thermal efficiency and a large power output been the primary objects of the work, they could have been achieved without undue difficulty although at considerably greater expense (the first

[37] See Appendix 11.

step in this direction being the purchase of an efficient industrial gas-turbine unit in place of a used, obsolete aircraft engine). Under these circumstances a budget of less than $700,000 for a 6-year program of design, construction and testing would have been completely impossible.[38]

Thus the output and efficiency figures obtained in this investigation must in no way be regarded as representative limits of the exhaust-heated cycle. The large cooling-water losses, for instance, which represent an irretrievable loss here, could be eliminated by letting the water evaporate and making use of the generated steam in a combined gas-steam cycle. The result would be a considerable increase in efficiency (26) (28b).

• Performance of Individual Components

1. Gas-turbine set

The second test series further demonstrated the reliability and low maintenance shown by the gas-turbine components when operating in the exhaust-heated cycle.

The only maintenance carried out on the experimental Dart aircraft unit was two washings of the compressor and inspection of the oil filters, which were found to be completely clean. When the engine had completed about 300 hours, some rubber oil seals adjacent to the turbine bearing had to be replaced because of overheating. At 875 hours a blade in the first-stage compressor failed and the spare Dart engine was then installed. These failures are attributed to the fact that the Dart engines used in the experiments were early obsolete models which had seen considerable service before being brought to McGill. The first engine had run at McGill for a longer period than is permitted for the latest engines in airline service. If these facts are borne in mind, the failures cannot be interpreted as evidence that similar failures are to be expected in industrial units.

2. Air-heater

The performance of the rebuilt heater conformed closely to the design predictions. The elimination of the parallel-flow section and the careful design of the baffle configuration reduced the shell-side pressure drop to one third of its original value.

The thermal ratio of the exchanger increased to 67 per cent mainly because of the elimination of the parallel-flow section but also because of the higher flow and hence higher coefficient on the gas side. Again, no deterioration of performance with time was noticeable, even in the longest test (Figure 47).

The most noticeable improvement occurred in the rate of tube-plate blockage. This was a direct result of the improvement of combustion performance and was also due to operation at lower temperatures. Figure 48 compares the rate of

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[38] It may be interesting to note here that a somewhat similar program of development of a 3,500-hp coal-burning turbine of 20 per cent thermal efficiency cost more than $8 million in the United States.

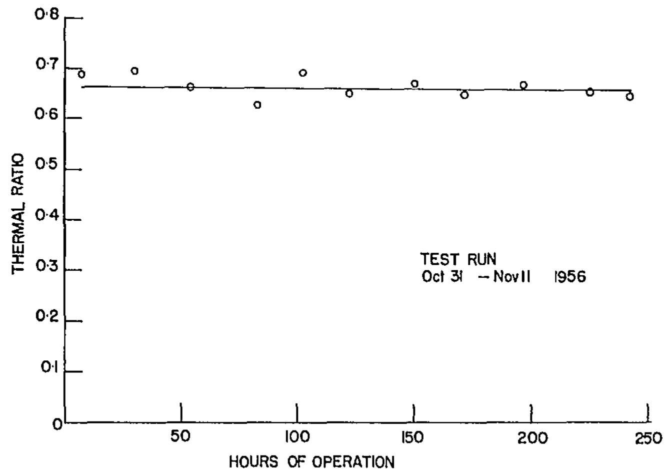

Figure 47. Air-heater thermal ratio.

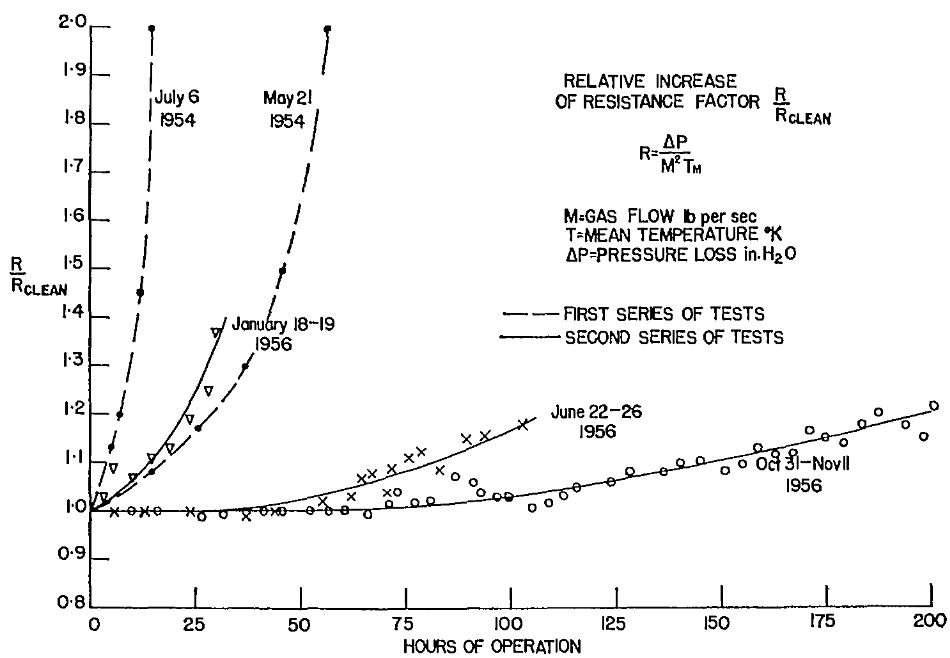

Figure 48. Heat-exchanger tube-plate blockage.

increase of the tube-blockage factor during the second series with the best results obtained in the 1954 tests. In the first series of runs, many of the tests had to be terminated when the resistance factor doubled in 10 to 20 hours. Towards the end of the second series, however, the increase was only about 15 per cent in more than 200 hours and was due partly to tube-plate blocking and partly to leakage produced by the corrosion of the tubes.

Mechanically, the heater performed satisfactorily although trouble was encountered with differential expansion of different alloy tubes. Depending on their expansion coefficients and on the temperature distribution around them, some tubes tended to pull out of the tube-sheet while others buckled in the bundle.

Originally these tubes were rolled into the hot tube-sheet only, and a slight clearance was provided at the other end. Uneven expansion caused binding anyway, and eventually all tubes were rolled into the cold tube plate.

The most important results from the heater operation related to tube materials and their resistance to corrosion. This is dealt with separately on page 94.

3. Furnace

(*a*) *General considerations*—The redesign of the furnace (described on page 71) proved highly successful, and only relatively minor modifications were required to obtain constant trouble-free operation.

The cyclone was normally operated at about stoichiometric air-fuel ratio, this condition giving a shorter flame and relatively little carryover. Actually, even when the air supply to the cyclone amounted to only 80 per cent of the stoichiometric requirement, no deterioration in combustion could be observed. Under these conditions, however, even a momentary increase in coal feed rate would result in an increase in the carryover of unburnt particles. Ordinarily, therefore, a slight excess of air was used.

Installation of the fully re-entrant throat improved combustion greatly, but the separation efficiency of the cyclone was still considerably affected by variations in the spray pattern of coal on the cyclone walls.

The injection of coal together with an equal amount of primary combustion air (which is a well-proven method in large cyclones) was unsatisfactory in this instance because the ignition delay was sufficient to cause the flame front to be established about 12 inches from the coal entry, or half-way down the cyclone. In large furnaces several feet long, the reaction would still be completed within the cyclone drum, but in this instance the flame would extend into the mixing zone, where it would be prematurely chilled by the dilution air. This type of operation would invariably result in an increased quantity of airborne particles in the exhaust gases.

It followed that, for satisfactory operation, the ignition delay had to be reduced and the flame front brought back close to the coal entry. This could be done by reducing the amount of air injected with the coal, although the entry was then more prone to blockage by coking and slag obstruction caused by local deficiencies of combustion air.

(*b*) *Coal entry*—Most of the early tests in this series were devoted to experimentation with various coal entries, and the swirl-cup type of entry described on page 76 and shown in Figure 41 was finally developed. From an investigation into the effect of the size of the entry on combustion, carryover, coke formation and wear, the following conclusions were drawn:

> Coke tends to form on any surface of the swirl cup which is not swept continuously clean by the coal and air or shielded from radiation by the dust cloud. The larger the diameter of the swirl cup from which the coal is discharged at a tangential velocity fixed by the size of the swirl port and the quantity of the transport air, the lower will be the carryover of airborne particles. The best cup size is therefore the largest that can be thoroughly swept by the coal and air stream.
>
> The wear of the swirl cup was overcome by the placing of a 2-inch strip of spring steel 0.02 inch thick inside the cup in the region of maximum wear. The strip was moved through glands at the rate of 1 inch every five hours. It wore up to 0.010 inch during passage through the cup, but the shape of the cup and the necessary spray pattern were maintained.

Figure 49.
Slag discharge from throat.

(c) *Cyclone throat*—At first it was found that most of the slag was washed through the throat into the mixing section instead of being drained through the slag-tap hole. From the wave formation frozen in the slag after a test, it appeared that the slag was blown by the air flow in liquid form along the surface of the cyclone and over the throat into the mixing section, where it solidified at the lower temperature, forming an extension of the conical shape of the throat (Figure 49).

The accumulation of slag also obstructed the discharge of the dilution air, causing poor temperature distribution in the furnace-outlet gases.

At the same time very little slag drained through the slag tap, but the absence of any accumulation of slag in the cyclone at the end of a run indicated that the flow of slag through the throat was not due to an excess of slag in the cyclone.

It was concluded that the slag flow was induced by the drag of the gas which followed the re-entrance of the throat and that a more pronounced re-entrance was required to separate the air flow from the wall and produce a reverse eddy in the re-entrant annulus which would keep the slag away from the gas outlet. This modification was incorporated before January 18 after about 10 hours of operation and raised the proportion of the slag percentage drained from a little more than zero to about 50 per cent (Figure 50).

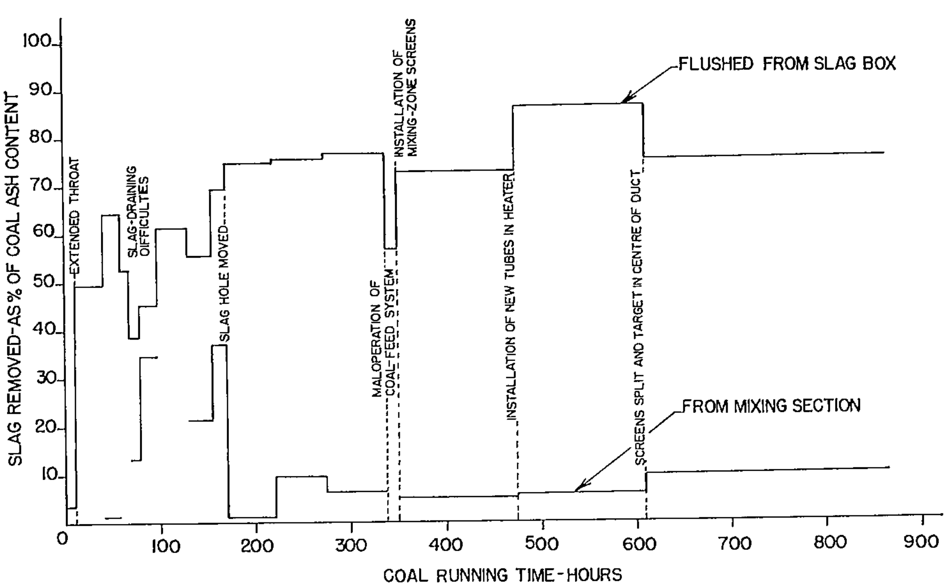

Figure 50. Slag removal (second series of tests).

Another successful modification was the installation of compressed-air jets just below the throat to force the frozen lumps of slag farther down into the mixing section, where they would not interfere with the dilution air.

Before the test run commencing October 31, a circular target was mounted outside the throat on the center line of the furnace to trap carryover which escaped through the throat. The great reduction that resulted in the rate of fouling of the screens and tube plate led to the conclusion that most of the sticky carryover particles leave the cyclone by this path at such a high axial velocity that they are projected upon the target, trapped by their own stickiness and solidified by the dilution air. Figure 51a shows the target on installation and Figure 51b shows the slag removed from the target after a run of 247 hours.

(*d*) *Slag-tap location*—After the addition of the more pronounced re-entrance to the cyclone throat, liquid slag accumulated at the point where the air flow was separated from the cyclone wall by the throat re-entrance and was carried up the circumference of the cyclone by the rotation of gases.

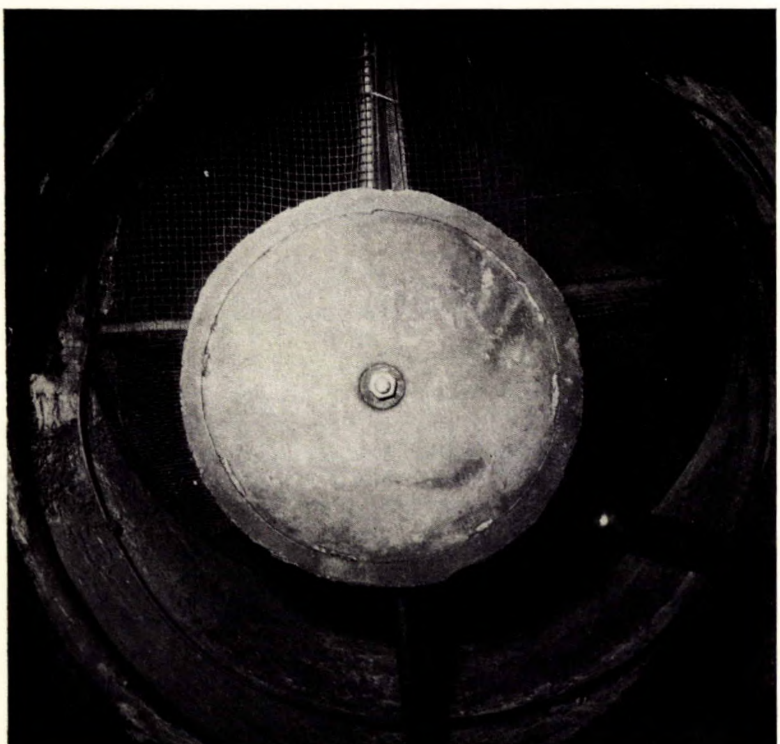

Figure 51a.
Target on installation.

Some of this slag could be seen to flow right over the top of the cyclone and down the opposite wall, thus making a complete revolution. Other portions of it would drip off the wall when near the top and either be blown through the throat or fall upon the throat cone and flow into the mixing zone.

The pattern of this flow, as well as a ridge of slag along the side of the cyclone displaced 24 degrees from the bottom by the tangential drag forces of the gases, was left in the frozen slag after shutting down (27).

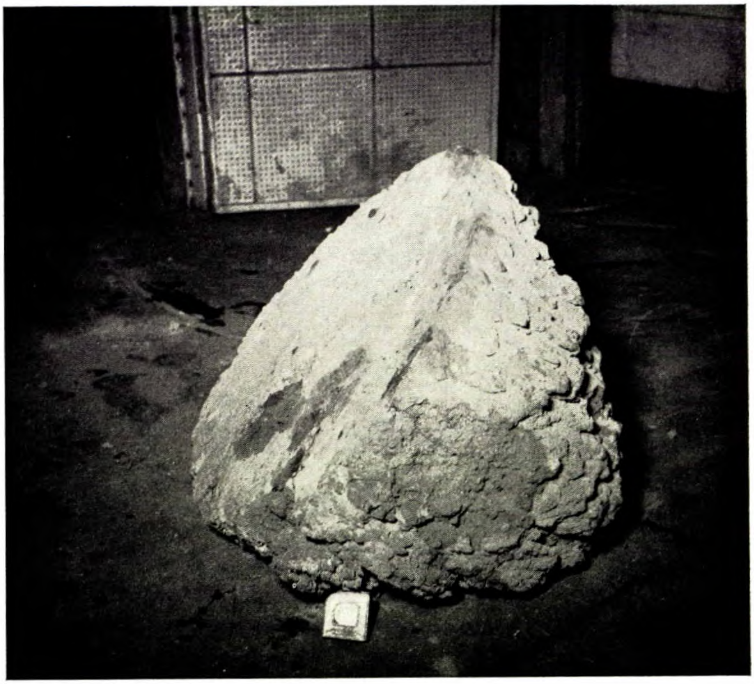

Figure 51b.
Slag accumulation from target after a 247-hour test.

If the cyclone was operated at a low tangential velocity, reasonable slag-tap flow could be maintained, but the vortex within the cyclone was then too weak to throw the coal particles to the walls and a large proportion of them passed through the throat and into the mixing section.

If too high a velocity was used, carryover was reduced but slag-tap flow stopped completely, and all the slag was carried through the throat in the manner described previously. Various systems of airbreaks were tried to allow the slag to flow against the air flow, but without success.

The slag box and cyclone were then modified to relocate the tap hole at the intersection of the longitudinal and circumferential slag ridges. This proved to be the solution to the problem, as the proportion of the slag drained from the cyclone increased to about 75 per cent of the total ash content of the coal, while the amount collected from the mixing section was reduced to less than 10 per cent (Figure 50).

(*e*) *Slag-removal gear*—At first the slag-removal gear consisted of a manually operated rotary slag-breaker below the slag drain, a hydraulic slag-classifier to divert large lumps into a catch pot, and a hydraulic ejector to eject the smaller pieces.

It was found necessary to operate the manual breaker almost continuously during times of poor combustion performance, and so mechanical operation was arranged.

The classifier, which was developed by rig tests with cold slag, was found to be unsatisfactory with hot slag owing to the buoyancy imparted to the slag by steam bubbles during cooling. It was replaced by a simple collecting funnel and hydraulic ejector with provision for back-flushing and poking to break up large pieces or divert them to the slag box.

In order to obtain early warning of blockage and permit rectification, a photoelectric slag-flow indicator and a mechanical-transport water-flow indicator were later incorporated. Thereafter there were no enforced stops caused by slag-removal difficulties.

(*f*) *Refractories*—The refractories in the mixing section of the redesigned furnace were completely reliable. They operated at a maximum temperature of about 1,350°F owing to effective air-cooling and good distribution of dilution air.

The refractory lining in the cyclone consisted of solidified slag after a few hours of operation and never presented a problem.

• Corrosion of Heater Tubes

Corrosion of heater tubes has proved to be the main problem encountered in the successful development of the heat exchanger.

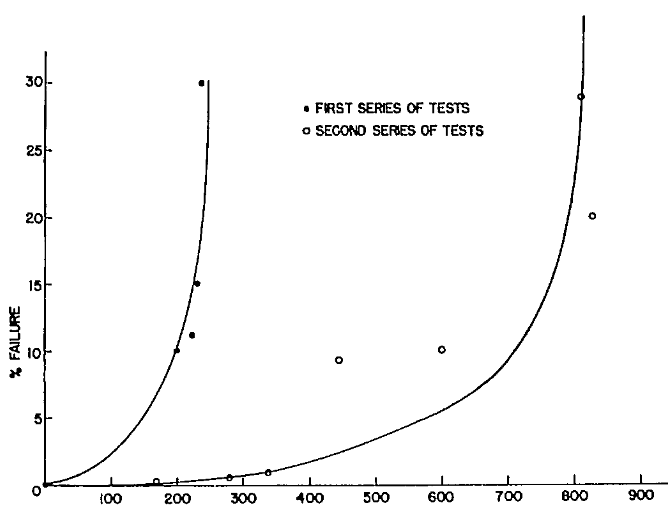

Figure 52. Cumulative percentage failure of Nimonic 75 tubes.

When the exchanger was first designed, Nimonic 75 tubes were accepted for reasons of availability although it was known that their high nickel content would make them more susceptible to sulphur attack. It was believed, however, that the large quantity of excess air present in the gas would provide an oxidizing atmosphere, which, it was hoped, would minimize the corrosion problem, as it was known that the attack tended to be serious only under reducing conditions.

In actual operation, however, the first failure of Nimonic 75 tubes occurred after about 200 hours, and the rate of failure became serious at about 250 hours of operation during the first series of tests (Figure 52).

The operating temperatures during this period were rather high (in excess of 2,000°F gas-inlet temperature), and the exchanger was operated for more than two hours at a gas temperature of about 2,130°F. With allowance for tube-plate cooling, the maximum gas temperature surrounding the tubes was of the order of 2,080°F and could reach 2,200°F locally in regions of less effective cooling.

The maximum tube temperature (calculated on the basis of average heat-transfer coefficients) would under these conditions reach 1,740°F, but locally (in areas of low air velocities on the shell side) it could exceed 1,900°F. Even these temperatures were on two occasions exceeded considerably for a few minutes owing to coal-feed troubles. As it was impossible to measure these excessive temperatures (they were beyond the instrument range of 2,220°F), it is not known exactly how hot the tubes became.

At these temperatures, however, all the coal ash reaching the heater would be in a partly fused state and would readily stick to tube surfaces.

In the region of these deposits the corrosive attack on the tube wall was most serious and resulted in complete perforation of the Nimonic 75 tubes. To gain an insight into the mechanism of the corrosive attack and to determine those factors which control it, an extensive investigation of the problem was undertaken by the Department of Mines and Technical Surveys in Ottawa.[39]

The work was divided into four phases:

(*a*) determination of the corrosive constituents in the coal samples as well as in ash and slag removed from the heat exchanger and other locations. (In addition, this team conducted routine coal and ash analyses to determine a heat balance for each test run of the experimental plant.)

(*b*) identification of constituents of ash or slag and of products of corrosion scraped from various heat-exchanger tubes, and determination of the mechanism of corrosive attack

[39] A special corrosion panel was organized at the Mines Branch. Four teams of engineers and scientists from the Physical Metallurgy, Fuel and Mineral Dressing and Process Metallurgy divisions conducted investigations on this problem. A full report on this work can be found in Reference 28a.

(*c*) comparison of the relative resistance to corrosion of various high-temperature alloys under conditions resembling those in the experimental plant

(*d*) metallographic examinations of tubes of various alloys removed from the heat exchanger after 481 hours of operation and after the termination of the tests.

The foregoing work resulted in the determination of the mechanism of corrosive attack and gave important information on the relative corrosion resistance of various alloys, thus indicating which of them showed greatest promise in this type of service.

1. Mechanism of corrosive attack

The investigation into the mechanism of corrosive attack yielded the following conclusions:

(*a*) The attack on the heat exchanger tubes was not caused by the slag alone.

Experiments with slag-covered Nimonic 75 tube samples showed no sign of attack at temperatures up to 2,150°F (1,175°C), or about 300 to 400°F above the temperature levels at which corrosion occurred in the heat exchanger.

(*b*) Presence in the slag even of small quantities of sulphur-bearing coal caused attack after only a few hours at temperatures in excess of 2,050°F. Slag coatings with a 20-per-cent or larger coal content caused visible attack after 24 hours at temperatures as low as 1,470°F (800°C). When the coal in the slag-coal mixture was replaced by pure carbon, no evidence of attack was observed. This was definite proof that a non-carbonaceous constituent of the coal was the cause of corrosion. Examination of the corroded sections showed the presence of a fusible mixture of metal and sulphides, thus definitely identifying sulphur as the coal constituent that was causing corrosion.

The average sulphur content of the coal burnt in the experimental plant was about 3½ per cent, the proportion in individual samples being as high as 5.3 per cent.

These findings made it possible to postulate the mechanism of corrosive attack: the presence of burning carbon in the slag deposited on the tubes created localized reducing conditions underneath lumps of partially burned coal. These conditions made the sulphur attack on the Nimonic 75 alloy rapid and severe, at temperatures of about 1,470°F (800°C).

In the light of these findings it becomes significant that the samples of ash scraped from the heat exchanger tubes contained usually no more than about 10 per cent of carbon, while the fly-ash collected by the sampling tube directly in front of the exchanger had a carbon content of 50 to 70 per cent. The difference was undoubtedly due to the fact that combustion of coal particles continued on the surface of the tubes, thus creating conditions favorable to attack. The coal at this temperature was in a plastic condition and softened sufficiently to spread slightly and adhere firmly to the metal. The areas of the most severe corrosion, leading to complete perforation of the tubes, invariably corresponded to places in which the coal-bearing slag was deposited.

The results of this investigation, which became known at the end of the first test series, showed that it was imperative to reduce the amount of ash deposited on the heat exchanger tubes in order to insure a satisfactory tube life. This could be done by reducing the furnace-ash carryover and insuring that the exchanger-inlet temperature was safely below the ash-fusion temperature.

Both these conditions were dependent on the satisfactory performance of the furnace, and the concerted effort applied towards the solution of this problem resulted in considerable improvement in tube life in the second test series. This is shown graphically in Figure 52, where the rate of Nimonic 75 tube failure is plotted against the hours of operation on coal. Thus, even with Nimonic 75 tubes, the failure rate was decreased by about 75 per cent in the second test series, and subsequent examination showed that alloys other than Nimonic 75 withstood the service conditions much more satisfactorily, no failures occurring in nearly 900 hours of operation.

2. Relative corrosion resistance of various alloys

(*a*) *Tests on tubes removed from heater*—A direct comparison of the corrosion resistances of various alloys was obtained both in the laboratory and, under actual operating conditions, in the experimental plant.

When at the end of the first series it became apparent that Nimonic 75 tubes were not very suitable for the prevailing service conditions, tubes of various alloys were incorporated in the exchanger. The various alloys tested are listed in Table VII,[40] which also shows their performance in 869 hours of service.

Table VII lists three types of failures that occurred in the experimental plant. The corrosion failures are the only ones of real interest here. The collapse of the 410 and 442 tubes was attributed to the fact that a wall thickness of .035 inch was not sufficient to withstand the external pressure at the prevailing temperature, at which the mechanical strength of these alloys is considerably reduced. Since heavier-gauge tubes could not be obtained, the thin-wall tubes were installed even though it was expected that they might not withstand the operating conditions.

As mentioned previously, the employment of various alloys in the bundle created differential-expansion troubles. In the worst cases (those concerning 304, 321, 347 and 316 tubes) this led either to severe buckling of the tubes or to their being pulled out of the tube-sheet. In either case the tubes were rendered unserviceable and it became necessary to withdraw them.

The corrosion failures were confined largely to Nimonic 75 tubes, although 304, 316, 321 and 347 steels also showed susceptibility. The tubes that failed by

[40] The composition of these alloys and the location of the tubes in the bundle are shown in Appendix 9 and Reference 28a.

Table VII

Tubes Installed in the Rebuilt Air-heater (November 1955)

Tube Material	Wall Thickness	Number of Tubes Installed	Number of Tubes Failed[41]		
			A	B	C
Type 304 stainless	.035″	6	1	0	5
Type 310 stainless	.035″	6	0	0	0
Type 316 stainless	.035″	4	1	0	1
Type 321 stainless	.035″	11	3	0	6
Type 347 stainless	.035″	6	1	0	2
Type 410 stainless	.035″	5	0	5	0
Type 442 stainless	.035″	14	0	9	0
Type 446 stainless	.064″	3	0	0	0
Incoloy	.064″	5	0	0	0
F.C.B.(T) (347)	.064″	5	0	0	0
Immaculate 5 (309)	.064″	5	0	0	0
Nimonic 75 (new)	.035″	6	136	0	0
Nimonic 75 (welded)	.035″	424			
Total		500			

[41] Column A lists tubes that failed by corrosion.
Column B lists tubes that collapsed owing to excessive loss of strength at high temperature.
Column C lists tubes that had to be removed owing to differential expansion troubles.

corrosion were distributed with fair uniformity throughout the bundle, and it was impossible to correlate their position with temperature differences existing at the intake. Detailed traverses established that these temperature differences, whose maximum was about 90°F (50°C), were not excessive.

Figure 53 shows the distribution of the mean gas temperatures at the heater inlet. The mean temperature for all the 1956 tests was 1,510°F (821°C), with a local maximum of 1,550°F (844°C).

Variation of temperatures with time is shown in Figure 54. The maximum temperature reached during the second series of tests was about 1,700°F (925°C), but this was of very short duration, and about 80 per cent of the running was done between 1,450 and 1,580°F.

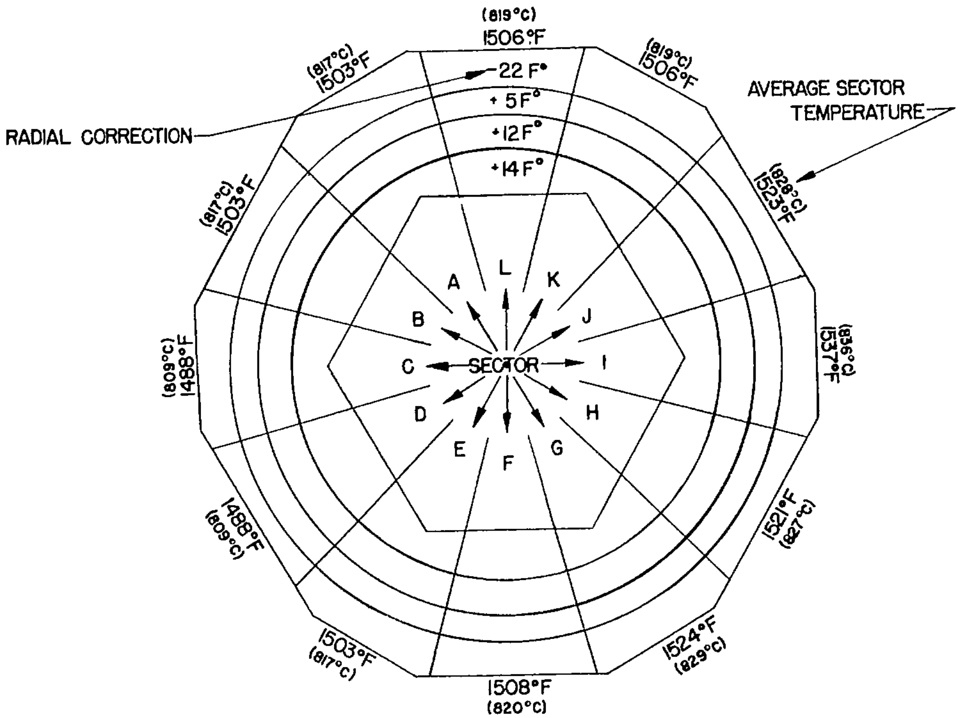

Figure 53. Gas-temperature distribution at heater inlet (second series of tests).

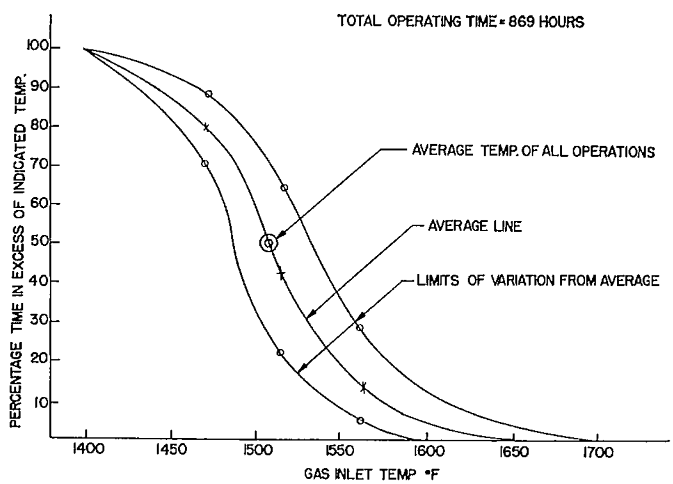

Figure 54. Heat-exchanger gas-inlet temperatures (second series of tests).

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The detailed knowledge of average temperatures made possible a fairly close estimation of the average tube temperatures. When these were being computed, the tube-sheet cooling-water losses were taken into account and the average heat-transfer coefficients were used. The following expression was derived[42] for the variation of tube-wall temperature with distance from the hot-end tube-sheet:

$t_w = t_{50}\,(0.8282-0.0360\,x)+5.722\,x+23$

where t_w = calculated tube-wall temperature (°C),

t_{50} = measured average gas temperature (°C) including correction factor for location (Figure 53),

x = distance (in feet) measured from a point 5 inches downstream from the front face of the tube-sheet.

It must be borne in mind that this formula was obtained by using average values of heat-transfer coefficients corresponding to average flow conditions prevailing during the tests.

While the coefficient for flow inside tubes can be calculated accurately and should not vary excessively from tube to tube (unless partial or complete blockage occurs), the same cannot be said of the shell-side coefficient. Here, particularly in the outlet-nozzle zone close to the tube-sheet, the flow pattern is unpredictable and the air velocity may vary considerably, thus producing significant differences in the local heat-transfer rates. Thus it is possible that some tubes were exposed to considerably higher temperatures than the average values obtained by the formula.

In spite of these limitations, the formula gives a useful comparison of the temperature levels at various distances from the tube-sheet. Table VIII[43] gives the average temperatures of the alloy tubes which did not fail in service.

Table VIII

Mean Tube Temperatures
(Second Series of Tests)

Mean gas temperature	1,519°F (827°C) ± 15° F
Maximum gas temperature	1,696° F (925°C)
Time at or above 1,516° F (825°C)	46%
Mean tube temperature at x=2″	1,303° F (707°C)
x=12″	1,267° F (687°C)
x=72″	1,045° F (563°C)

As shown by Table VII, a number of alloys withstood 869 hours of operation at these temperatures without failure. These alloys were dispersed throughout the exchanger as indicated in Appendix 9 and thus were located in areas where the adjoining Nimonic 75 tubes showed numerous failures.

[42] See Appendix 12.

[43] Based on Reference 28, Table II, page 195.

At the end of 481 hours of testing, and again at the end of the test series (869 hrs.), several tubes were removed for examination. They were samples of the 11 different materials listed in Table VII and fell, roughly, into the following four categories:

(*a*) Nimonic 75 (a nickel-base alloy with a high chromium content)

(*b*) Types 446, 442 and 410 (ferritic, corrosion- and oxidation-resistant alloys with decreasing amounts of chromium, as listed)

(*c*) Types 304, 316, 321 and 347 (austenitic, corrosion- and oxidation-resistant steels of the group containing 18 per cent chromium and 8 per cent nickel)
Types 309 and 310 (which have a slightly higher percentage of both chromium and nickel)

(*d*) Incoloy (a new alloy containing about 21 per cent chromium and 32 per cent nickel).

The last three alloys were tested only after 869 hours of operation.

The samples of each material examined were 1 foot in length and were taken from each end and from locations 6 feet from each end. Examination was concentrated on the samples from the hot end, which was the more severely attacked.

The metallurgical examinations comprised:

(*a*) macroexamination of the external characteristics of the scales and scaled surface

(*b*) measurement of mean and maximum loss of thickness

(*c*) metallographic examination of the scaled inner surfaces in cross-section

(*d*) metallographic examination of structural changes.

The initial examination (after 481 hours) showed that Type 446 was considerably superior to other alloys. It was the only material on which scale formed uniformly without significant selective attack at grain boundaries or by pitting.

Its superiority to Nimonic 75 is indicated in Figure 55. The loss-of-thickness measurements also indicated the superiority of Type 446, except for a few inches at the hot end, where Type 442 showed the smallest loss. Examination of new samples after 869 hours of service confirmed the superiority of Type 446 over the materials examined after 481 hours, but showed that Incoloy and Types 309 and 310 steels, which were not available for the initial examination, gave even better results. The rate of scale build-up on Type 446 tubes increased considerably in the last 388 hours, as illustrated in Figure 56. The appearance of the scale was also changed, the coating having become much rougher (Figure 57).

The materials which appeared most promising after 869 hours were Incoloy, Type 309 and Type 310, although the last two showed local pitting that indicated highly localized attack on the metal (Figure 58). (Susceptibility to pitting is an undesirable characteristic.) The scale on the Incoloy samples was fairly uniform (Figure 59), and there was no evidence of pitting, the areas of general attack

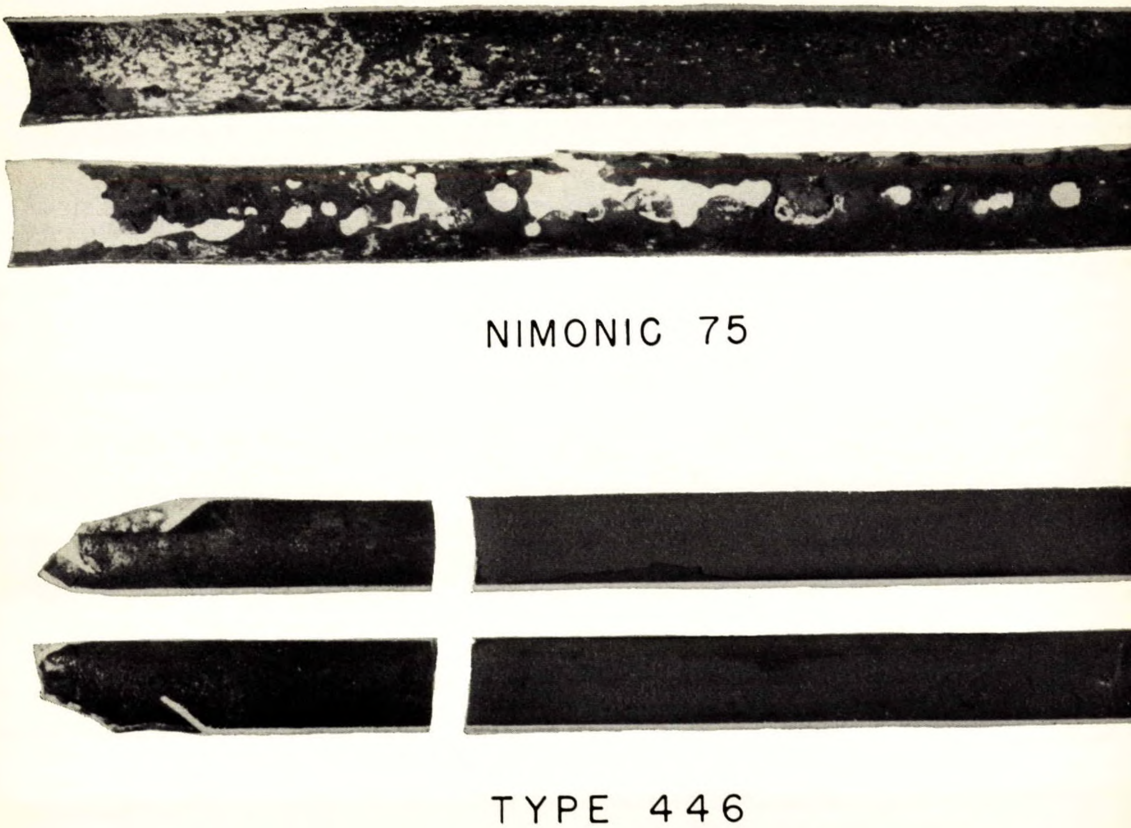

Figure 55. Comparison of Nimonic 75 and Type 446 tubes after 481 hours of operation (approximately one-half actual size).

Figure 56. Scale build-up on Type 446 tube (x16) after 481 hours and 869 hours (typical cross-sections of scale; scale removed about 15 inches from tube insert).

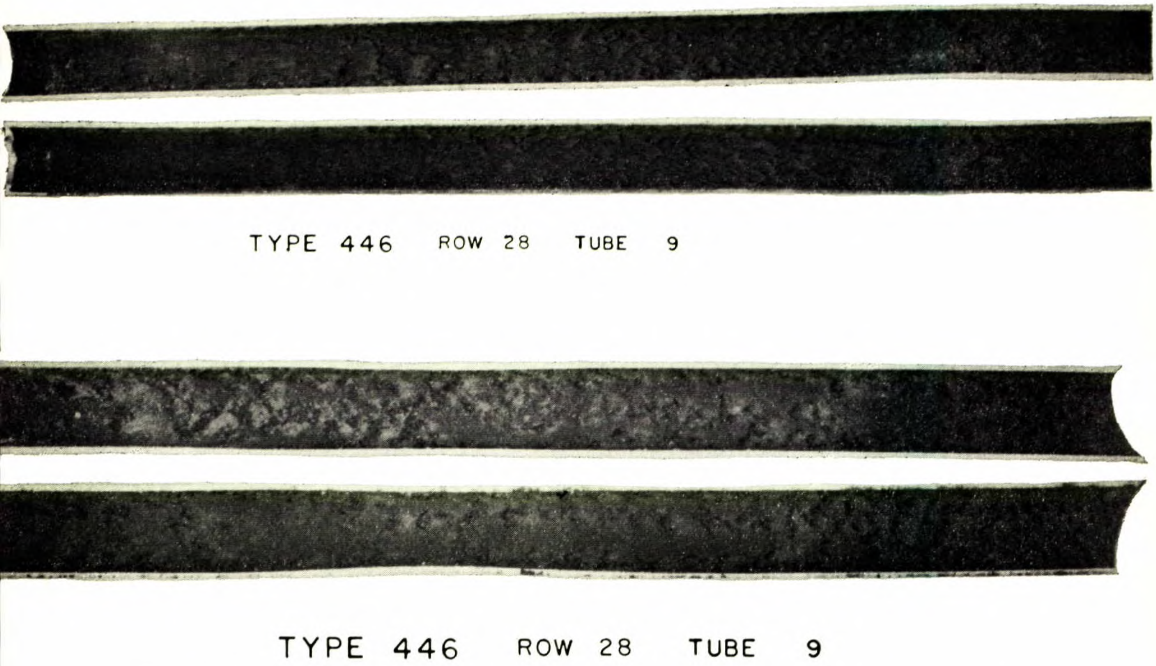

Figure 57. Scale deposit on Type 446 tube (inner surfaces approximately one-half actual size).

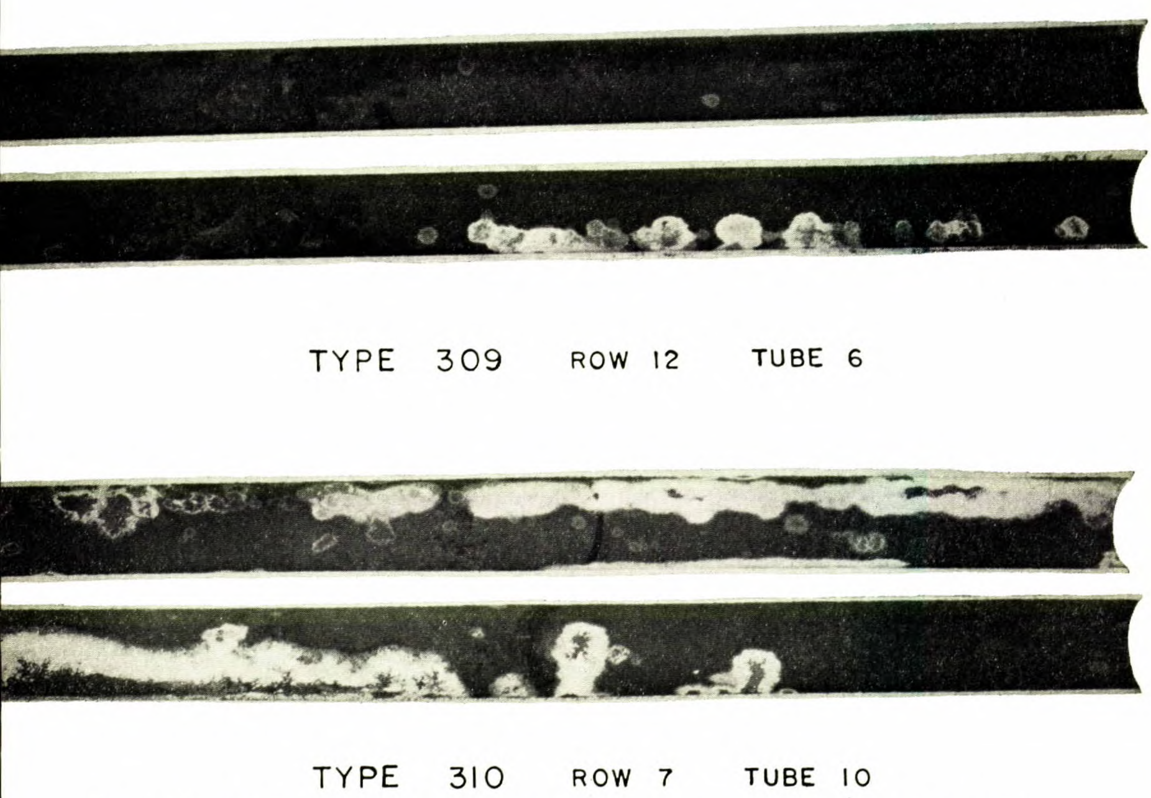

Figure 58. Scale deposit on Types 309 and 310 tubes (inner surfaces approximately one-half actual size.)

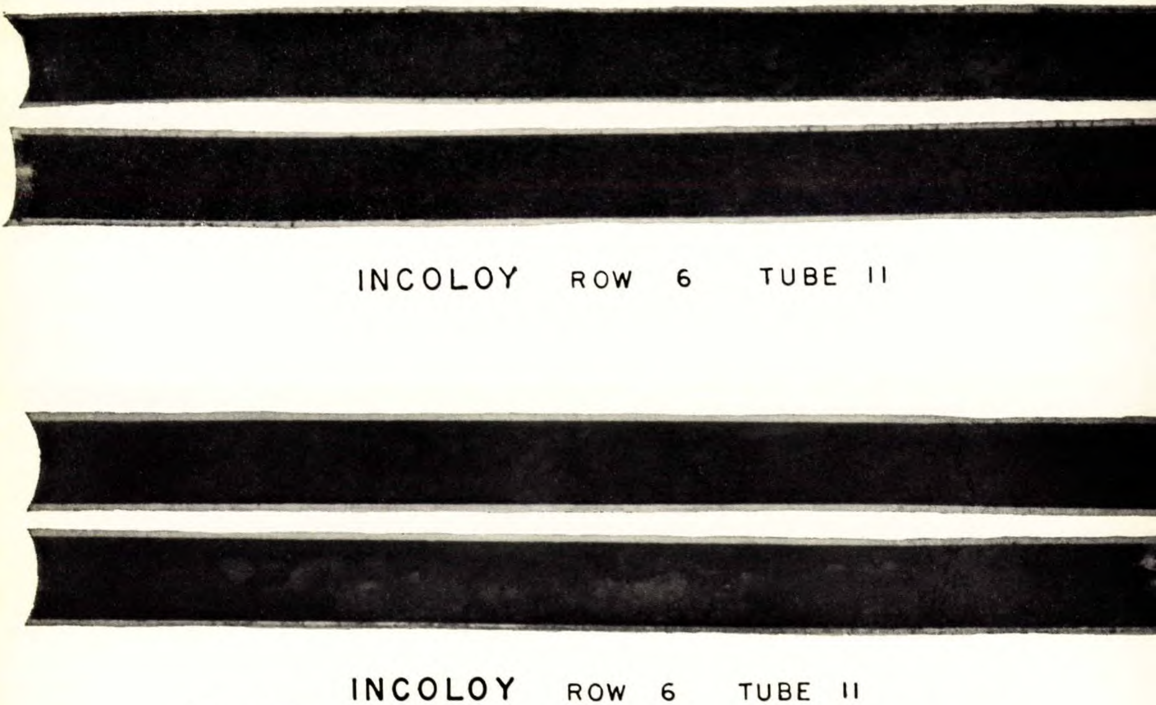

Figure 59. Scale deposit on Incoloy tube (inner surfaces approximately one-half actual size).

Figure 60a. Corrosion attack on Incoloy (row 21, tube 11 (x50), 6 feet from ceramic insert).

This is the most severely attacked area observed in any of the Incoloy tubes. The wall thickness has been reduced to 0.0578 inch. The wall thickness of the sample of new tubing ranged from 0.0692 to 0.0624 inch with a mean of 0.0666 inch. Since the minimum of 0.0578 inch occurred on the thinner half of the tube, the metal loss was estimated to be in the range of 7.4 to 13.2 per cent.

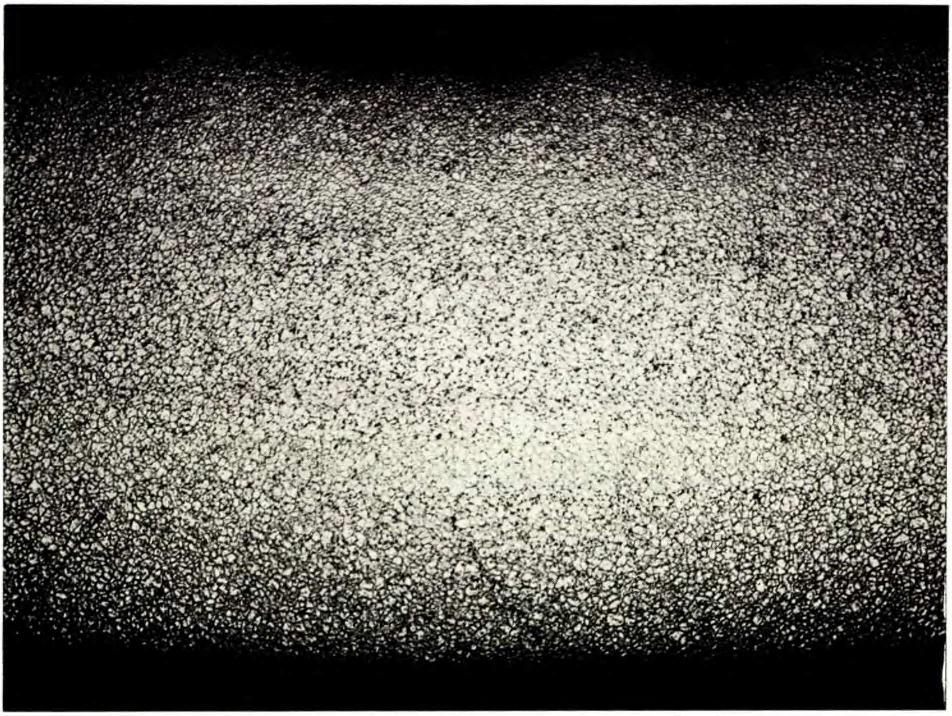

Figure 60b. Corrosion attack on Type 309 (row 12, tube 6 (x50), 1 inch from ceramic insert).
This is the most severely pitted area observed in the Type 309 tube. The wall thickness has been reduced to 0.0620 inch. The minimum individual thickness of 0.0588 inch, which also occurred in this section, was estimated to represent a metal loss in the range of 10.2 to 11.8 per cent.

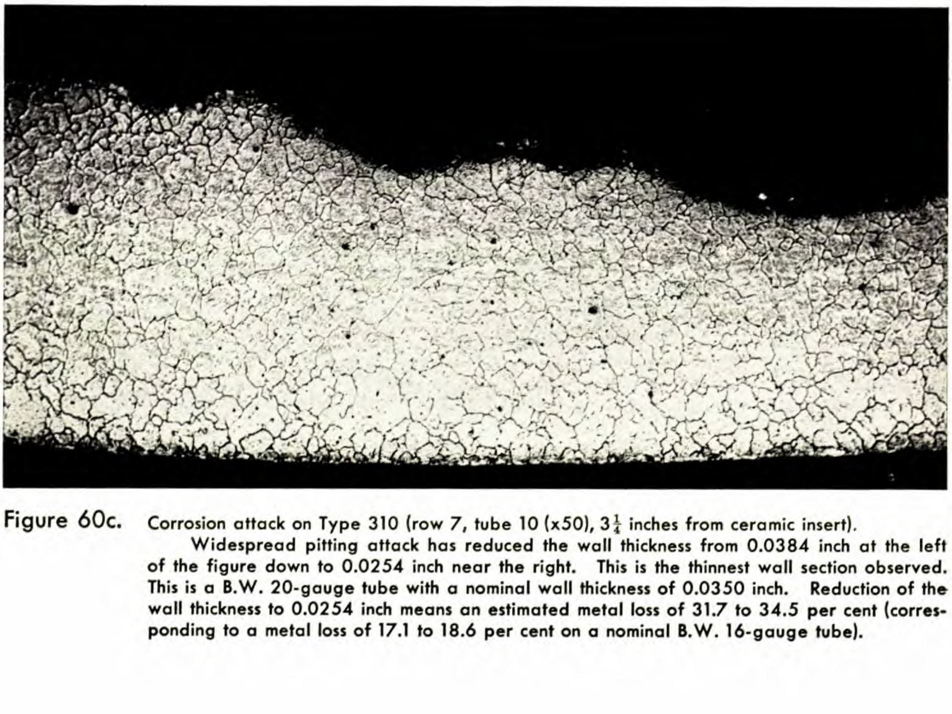

Figure 60c. Corrosion attack on Type 310 (row 7, tube 10 (x50), $3\frac{1}{4}$ inches from ceramic insert).
Widespread pitting attack has reduced the wall thickness from 0.0384 inch at the left of the figure down to 0.0254 inch near the right. This is the thinnest wall section observed. This is a B.W. 20-gauge tube with a nominal wall thickness of 0.0350 inch. Reduction of the wall thickness to 0.0254 inch means an estimated metal loss of 31.7 to 34.5 per cent (corresponding to a metal loss of 17.1 to 18.6 per cent on a nominal B.W. 16-gauge tube).

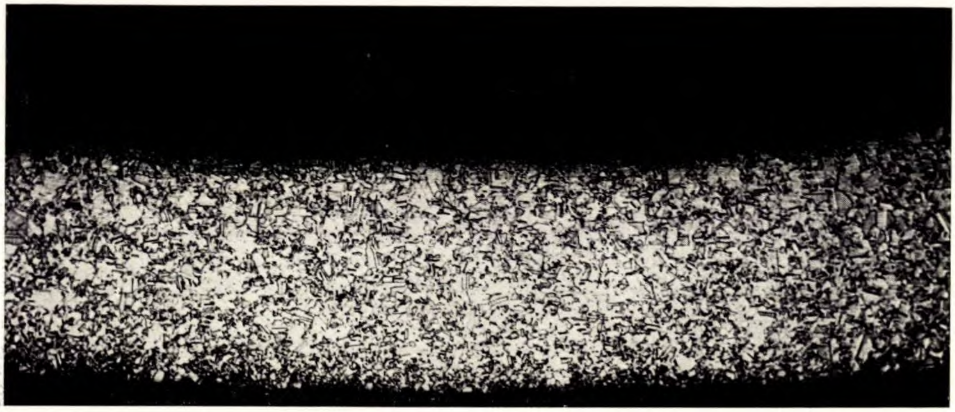

Figure 60d. Corrosion attack on Type 347 (row 5, tube 11 (x50), 6 inches from ceramic insert). Severe general attack has reduced the wall thickness to 0.0248 inch. This is a B.W. 20-gauge tube with a nominal wall thickness of 0.0350 inch.

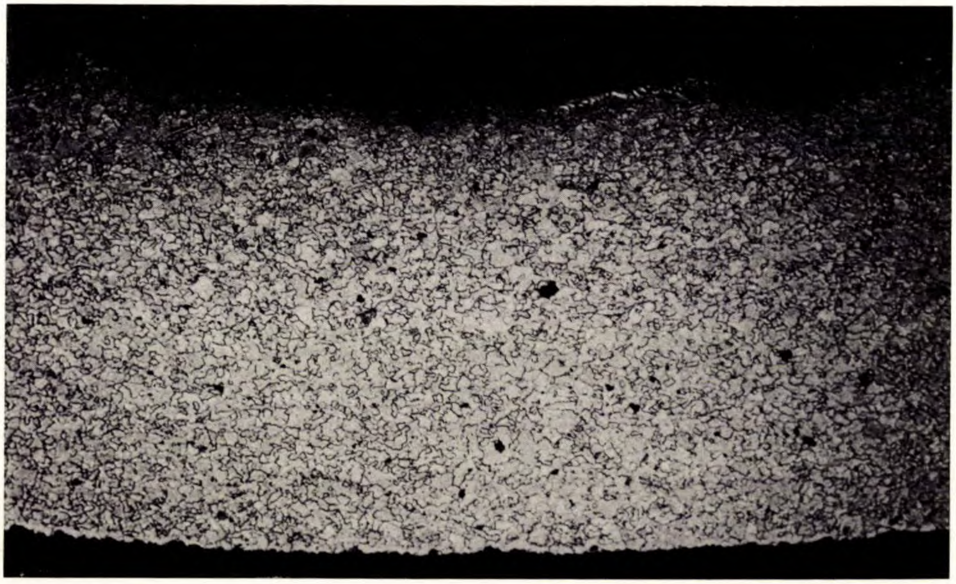

Figure 60e. Corrosion attack on Type 446 (row 28, tube 9 (x50), $1\frac{7}{8}$ inches from ceramic insert). Severely attacked area where the wall thickness has been reduced to 0.0464 inch. This is a B.W. 16-gauge tube with a nominal wall thickness of 0.0650 inch.

showing a uniform loss of metal. Figures 60, (*a*) to (*e*), illustrate the appearance of the various tubes in areas of the most severe attack.

On the basis of loss-of-thickness measurement Incoloy and Types 309 and 310 steels proved superior to Type 446 after 869 hours. Table IX[44] gives a summary of the results.

[44] Based on Reference 28a, Table IV (page 198).

Table IX

Mean Percentage Loss of Metal after 869 Hours of Operation

(Note: A loss of 1.5 per cent is equivalent to a loss of 0.001 inch of wall thickness of B & W 16-gauge tubes.)

Material of Tube	Mean Thickness, New (inches)	Average Percentage Loss (distance from ceramic insert)			
		1″-2″	3″-12″	18″-30″	48″-72″
Incoloy (4 tubes)	.067	1.0	0.9	0.6	0.8
Type 309	.067	0.3	0.4	0.1	0.4
Type 310	.039	1.5	1.7	0.0	0.2
Type 347 (2 tubes)	.068 (.037)	11.5	14.2	8.0	—
Type 446 (2 tubes)	.067	15.3	6.1	10.8	—

(*b*) *Laboratory tests on alloy samples*—In connection with the corrosion investigation, a test rig was constructed at the Mines Branch laboratory, where samples of alloys could be subjected to conditions resembling those of the experimental plant. The samples were heated in a small horizontal tube while small pieces of coal were dropped at a controlled rate on the surface of the samples. Preheated air was blown through the tube, and thus actual operating conditions in the experimental plant were simulated.

These tests showed again that both Incoloy and Type 446 were superior to Type 442, Type 347 and Nimonic 75. But failure, after 50 hours of testing, to observe any difference between the relative resistance of Incoloy and that of Type 446 seemed to contradict the results obtained in the experimental plant.

This could be due to the fact that the conditions under which the laboratory tests were carried out were typical of those prevailing during the early period of plant operation, when heavy deposits of burning coal particles on the tube walls created conditions particularly favorable to sulphidation attack on nickel-bearing alloys.

The considerable improvement in combustion performance in the second series of tests, which must be regarded as more representative of the plant potential, brought an appreciable change in conditions. This may account for the apparent disparity between the results of the laboratory tests and those of service in the heater.

Limitation of the testing time to 50 hours also made it impossible to draw definite conclusions. Further tests must be conducted before it can be definitely ascertained which of the alloys is best suited to this type of service.

CONCLUSIONS

Those who conducted the experiments at the Gas Dynamics Laboratory under the sponsorship of the Mines Branch inscribed an important first for Canadian engineering in the annals of power generation. Their work resulted in the first successful run of a coal-fired, exhaust-heated gas turbine. This achievement was widely recognized throughout the world, as evidenced by articles in engineering periodicals on both sides of the Iron Curtain, and by the tremendous interest shown in the Conference on Coal-burning Gas Turbines, held in November 1956 at McGill University and attended by some 90 delegates representing countries ranging from Canada to South Africa and Australia.

From a technical viewpoint, the experimental work must be considered highly successful. At a cost amounting to less than 8 per cent of that of a similar project conducted outside Canada, it was demonstrated that the exhaust-heated cycle is feasible for burning low-grade coal, and that the exhaust-heated turbine is a rugged machine that can withstand considerable abuse and maloperation with only a gradual loss of performance. In 1,000 hours of operation, the reliability of the experimental plant was improved to such an extent that 200 hours of non-stop testing was achieved. This represented by far the longest non-stop run of any coal-fired gas turbine in the world.

There were four main reasons why the experimental plant did not achieve the power and efficiency predicted:

(*a*) large cooling-water losses in the cyclone furnace and heat-exchanger tube-sheet

(*b*) large heat losses caused by the exposure of surfaces in the spread-out arrangement required in the test plant for easy access to components

(*c*) the use of a low-efficiency aircraft gas turbine

(*d*) limitations on the heater-inlet gas temperature made necessary by ash-deposition considerations.

The first three reasons can be easily eliminated in an improved design. The cooling-water losses can be considerably reduced by adopting an air-tube heater (without a water-cooled tube-sheet) and using a different furnace design,[45] or they can be eliminated by utilizing the steam for additional power generation or for process purposes. The heat losses from exposed surfaces can be greatly reduced in a compact design, and the substitution of an efficient industrial turbine would further raise the efficiency.

The problem of the maximum temperature that can be allowed at the heater inlet is linked directly to the performance of the furnace. It was proved definitely that corrosion of the heater tubes was due to the presence of sulphur-bearing coal in the ash deposited on the tubes. If complete combustion can be achieved, the temperature levels can be increased considerably since fly-ash alone does not

[45] A comparison of various furnace designs can be found in Reference 28, page 85.

cause corrosion. Reduction in the size of ash particles would also facilitate complete combustion. The work performed already has gone a long way towards the solution of these problems.

The search for the most suitable heat-exchanger materials has uncovered promising possibilities, but further testing is needed to establish the most satisfactory alloy. This can be done only by subjecting various samples to simulated (or actual) operating conditions for periods of several thousand hours. The results of 50-hour (or even 800-hour) tests cannot be extrapolated satisfactorily to 20,000 hours.

The work described in this report is limited to the design and testing of the experimental plant. Concurrently with the experimental work, however, a number of studies were made of the economic possibilities of this type of engine and its possible industrial applications (26, 28b). Although the exhaust-heated turbine could not reach the efficiency of, say, the diesel engine in locomotive application, it could still show sizable operational savings in line service owing to the large difference in the cost of fuel.

In power-plant application, the exhaust-heated cycle looks very attractive when combined with a steam plant. The greatest benefits are gained in stations of medium size, about 30 to 50 megawatts, with the gas turbine developing about 15 per cent of the power.

As always in the case of the development of a new type of engine, laboratory work alone cannot produce the final design. It is believed that the experiments which were conducted at McGill provide the necessary basis for continuation of the work on the coal-burning turbine.

The most serious problem encountered, that of corrosion, while not solved completely, was carried to a stage where a number of promising alloys were uncovered. Further testing under simulated (or, preferably, actual) operating conditions is required in order to confirm the results obtained so far.

The feasibility, operational flexibility and ruggedness of this type of engine have been fully demonstrated in laboratory operation. The economic possibilities have been demonstrated on paper only. They can be confirmed in practice only by the construction and operation of a full-sized prototype.

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Appendices

Appendix 1

THERMODYNAMIC-CYCLE CALCULATIONS

The calculations given below were based on the approximate performance data on the RD-A4 Dart engine, available at the time.

The operating conditions chosen were as follows:

Speed	13,500 rpm
Turbine-inlet temperature T_4[46]	1,340°F
Compressor mass flow	14.6 lb/sec
Compressor pressure ratio	4.35
Compressor efficiency	75%
Turbine efficiency	80.5%

COMPRESSOR

Inlet temperature	59° F (519°R)
Inlet pressure	14.7 psia

$$\text{Isentropic temperature rise} = \Delta T_{is} = T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

where $\gamma = 1.398$ within the temperature range encountered in the compressor

$$\Delta T_{is} = 519\,[4.35^{0.285} - 1] = 270°\text{F}$$

$$\text{Compression efficiency } \eta_c = 75\%$$

Hence the actual temperature rise in the compressor is

$$\Delta T_c = \frac{270}{.75} = 360°\text{F}$$

Hence the compressor-outlet temperature is

$$T_2 = 59 + 360 = 419°\text{F}$$

TURBINE

Inlet temperature	1,340°F (1,800°R)
Pressure drop between compressor and turbine	1.90 psi
Pressure drop between turbine and exhaust stack	1.18 psi

$$\text{Turbine-inlet pressure } P_4 = 14.7 \times 4.35 - 1.90 = 62.1 \text{ psia}$$

$$\text{Turbine-outlet pressure } P_5 = 14.7 + 1.18 = 15.88 \text{ psia}$$

$$\text{Turbine pressure ratio} = \frac{62.1}{15.88} = 3.91$$

$$\text{Isentropic temperature drop } \Delta T_{is} = T_4\left[1 - \frac{1}{(P_4/P_5)^{\gamma - 1/\gamma}}\right]$$

[46] Subscripts refer to location shown in Figure 6, on page 6 of the report.

The average specific heat of the air in the temperature range encountered in the turbine is 0.264 Btu/lb °F.

Hence
$$\gamma = \frac{C_p}{C_v} = \frac{C_p}{C_p - R} = 1.343$$
$$(R = 0.0685 \text{ Btu/lb °F})$$
$$\Delta T_{is} = 1{,}800 \left[1 - \frac{1}{3.91^{0.255}}\right] = 529 \text{ °F}$$
$$\text{Turbine efficiency } \eta_t = 80.5\%$$

Hence the actual temperature drop in the turbine is
$$\Delta T_t = 529 \times .805 = 426 \text{ °F}$$

Hence the turbine outlet temperature is
$$T_5 = 1340 - 426 = 914 \text{ °F}$$

HOT HEAT EXCHANGER

Gas flow in the exchanger........................ 8.0 lb/sec
Gas-inlet temperature T_6........................1,965°F
Gas-outlet temperature (to be equal to T_5)......... 914°F

The average specific heat of combustion gases in the temperature range encountered in the hot heat exchanger is 0.286 Btu/lb °F.

Heat lost by the gas in the exchanger is
$$Q_g = W\,C_{pg}\,(T_6 - T_{out}) = 8 \times .286\,(1965 - 914) = 2410 \text{ Btu/sec}$$

Air flow in the exchanger.......................... 14.6 lb/sec
Air-outlet temperature T_4.......................... 1,340°F

The average specific heat of the air in the temperature range encountered in the exchanger is 0.264 Btu/lb °F.

The air-temperature rise in the exchanger is
$$T_4 - T_3 = \frac{Q}{W_a C_{pa}} = \frac{2410}{14.6 \times .264} = 625\text{°F}$$

Hence the air temperature at the exchanger inlet is
$$T_3 = 1340 - 625 = 715\text{°F}$$

The heat-exchanger thermal ratio is
$$\frac{1340 - 715}{1965 - 715} = 0.5$$

COLD HEAT EXCHANGER

Gas and air flow in the exchanger (assumed equal)..14.6 lb/sec
Air-inlet temperature........................... 419°F
Air-outlet temperature.......................... 715°F
Gas-inlet temperature........................... 914°F

The average specific heat of the air in the temperature range encountered in the exchanger is 0.250 Btu/lb°F.

The heat transferred to the air in the exchanger is
$$Q = 14.6 \times 0.250 \times (715 - 419) = 1080 \text{ Btu/sec}$$

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The average specific heat of combustion products in the temperature range encountered in the exchanger is .262 Btu/lb °F.

Gas-temperature drop in the heat exchanger is

$$T_{in} - T_7 = \frac{1080}{14.6 \times .262} = 282\ ^\circ\mathrm{F}$$

Hence the gas-outlet temperature is

$$T_7 = 914 - 282 = 632\ ^\circ\mathrm{F}$$

The heat-exchanger thermal ratio is

$$\frac{715 - 419}{914 - 419} = 0.6$$

The over-all heat-exchanger thermal ratio is

$$\frac{T_4 - T_2}{T_6 - T_2} = \frac{1340 - 419}{1965 - 419} = 0.6$$

Appendix 2

DESIGN OF HOT HEAT EXCHANGER

1. THERMAL DESIGN

Derivation of the Exchanger Design Formulae

Nomenclature:

d_o	=	Tube diameter—ft
L	=	Length of tubes—ft
N	=	Number of tubes
p	=	Tube pitch $= S \times d_o$
U	=	Over-all coefficient of heat transfer—Btu/sec °F ft^2
Q	=	Total heat transferred—Btu/sec
h	=	Coefficient of heat transfer—Btu/sec °F ft^2
Cp	=	Specific heat—Btu/lb °F
ΔP	=	Pressure loss—lb/ft^2
μ	=	Fluid viscosity—lb/ft sec
ρ	=	Fluid density—lb/ft^3
M	=	Mass flow—lb/sec
ΔT_m	=	Logarithmic mean temperature difference—°F
G	=	Mass flow per unit of flow area—lb/hr ft^2

The subscripts i and o are used to denote the inside and the outside of the tube respectively, and $r = \dfrac{d_i}{d_o}$.

The basic formulae for heat exchange are:

$$Q = U.\ A_s\ .\ \Delta T_m \qquad [1]$$

$$\frac{1}{U} = \frac{d_o}{d_i}\frac{1}{h_i} + \frac{1}{h_o} \qquad [2]$$

$$h = 0.0293\ Cp\ \frac{\mu^{0.2}\ M^{0.8}}{A^{0.8}\ d^{0.2}} \qquad [3]$$

$$\Delta P = \frac{4L}{d}\ \frac{0.046}{R_e^{0.2}}\ \frac{\rho V^2}{2g} \qquad [4]$$

The coefficient of .0293 is obtained by using a Prandtl number of .67.

The pressure loss as given in Eq. [4] is the isothermal loss for flow inside tubes (or outside tubes with no appreciable cross-flow).

Assuming equiangular pitching of the tubes and pure longitudinal flow outside the tubes, Outside tubes:

$$\text{Total flow area } (A_o) = N\frac{\pi}{4}(1.1\,S^2 - 1)\,d_o^2$$

$$\text{Effective diameter } (d_{eo}) = (1.1\,S^2 - 1)\,d_o$$

$$\text{Surface } (A_s) = N\,\pi\,d_o\,L$$

Inside tubes: $(d_i = r\,d_o)$

$$\text{Total flow area } (A_i) = N\,\frac{\pi}{4}\,(r\,d_o)^2$$

$$\text{Effective diameter } (d_{ei}) = r\,d_o$$

$$\text{Surface}^{47}\ (A_s) = N\,\pi\,d_o\,L$$

$$\frac{h_i}{h_o} = \left(\frac{Cp_i}{Cp_o}\right)\cdot\left(\frac{\mu_i}{\mu_o}\right)^{0.2}\cdot\left(\frac{M_i}{M_o}\right)^{0.8}\cdot\left(\frac{A_o}{A_i}\right)^{0.8}\cdot\left(\frac{d_{eo}}{d_{ei}}\right)^{0.2}$$

Substituting values for the areas and effective diameters in the foregoing, and reducing gives:

$$\frac{h_i}{h_o} = \frac{1}{r^{0.8}}\,K'\,(1.1\,S^2 - 1)$$

where $K' = \left(\frac{Cp_i}{Cp_o}\right)\left(\frac{\mu_i}{\mu_o}\right)^{0.2}\left(\frac{M_i}{M_o}\right)^{0.8}$

$$h_i = \frac{0.0293\ Cp_i\ \mu_i^{0.2}\ M_i^{0.8}}{\left(N \times \frac{\pi}{4}\,r^2 d_o^2\right)^{0.8}\ (rd_o)^{0.2}} \quad \text{from [3]}$$

But $$\frac{1}{U} = \frac{1}{rh_i}\left[1 + \frac{rh_i}{h_o}\right]$$

$$= \frac{A_s}{Q}\,\Delta T_m$$

Therefore

$$UA_s = \frac{Q}{\Delta T_m} = \frac{0.0293\ (Cp_i\ \mu_i^{0.2}\ M_i^{0.8})\ N^{0.2}\ L\ \pi d_o r}{\left(\frac{\pi}{4}\right)^{0.8}(rd_o)^{1.8}\left[1 + \frac{1}{r^{0.8}}\,K'\,(1.1\,S^2 - 1)\right]} \quad\text{........[5]}$$

From [4]

$$\Delta P_i = \frac{0.0920\ M_i^{1.8}\ \mu_i^{0.2}}{g\ \rho_i} \times \frac{L}{\left(\frac{\pi}{4}N\right)^{1.8}\left(rd_o\right)^{4.8}}$$

which on rearranging, gives

$$L = \frac{g}{.092}\,\frac{\rho_i}{M_i^{1.8}\ \mu_i^{0.2}}\ \Delta P_i\left(N\frac{\pi}{4}\right)^{1.8}(rd_o)^{4.8}\quad\text{..................[6]}$$

---

[47] d_o is used since all heat transfer coefficients are referred to the outside diameter.

Combining [5] and [6] and reducing:

$$N^2 = \frac{Q}{\Delta T_m} \frac{1.275}{g} \frac{1}{r^4} \left[\frac{M_i^{1.8} \ \mu_i^{0.2}}{\rho_i}\right] \frac{1 + \frac{1}{r^{0.8}} K' \ (1.1\,S^2 - 1)}{\Delta P_i \ (Cp_i \ \mu_i^{0.2} \ M_i^{0.8}) \ d_o^4}$$

If the temperature of the fluids and the mass flow for which the heat exchanger is designed are known, the following quantities may be evaluated using thermodynamic tables (29):

$$K' = \frac{Cp_i}{Cp_o} \left(\frac{\mu_i}{\mu_o}\right)^{0.2} \left(\frac{M_i}{M_o}\right)^{0.8}$$

$$B = Cp_i \ \mu_i^{0.2} \ M_i^{0.8}$$

$$C = \frac{M_i^{1.8} \ \mu_i^{0.2}}{\rho_i}$$

$$D = \frac{M_o^{1.8} \ \mu_o^{0.2}}{\rho_o}$$

By using these quantities, the design formulae may be simplified to the following forms:

$$\text{I.} \ \frac{\Delta P_i}{\Delta P^0} = \frac{1}{r^{4.8}} \times \frac{C}{D} \ [1.1 \ S^2 - 1]^3$$

$$\text{II.} \ N^2 = 0.0396 \ \frac{1}{r^4} \times \frac{C}{B} \times \frac{Q}{\Delta T_m} \times \frac{1 + \frac{1}{r^{0.8}} K' \ [1.1 \ S^2 - 1]}{\Delta P_i \ d_o^4}$$

$$\text{III.} \ L = 228.9 \ r^{4.8} \ \frac{1}{C} \ N^{1.8} \ \Delta P_i \ d_o^{4.8}$$

The foregoing formulae were used to obtain the size of the exchangers once the operating temperatures became fixed.

This required a number of trial-and-error calculations, particularly in the case of the hot exchanger, where the transfer point from counter to parallel flow had to satisfy definite pressure-drop, heat-transfer and tube-metal temperature requirements.

2. TUBE SHEET DESIGN[48]

The component of the hot heat exchanger exposed to the highest temperatures is the hot-end tube plate. In view of the elevated temperatures, the knowledge of the temperature distribution in the tube plates was vital. The technique developed to obtain this information and the postulates on which it is based will now be outlined.

Temperature Analysis

It was assumed that a certain section of the tube plate could be associated with each tube. This section only would be affected by heat flow from its particular tube, as shown in Figure A2-1. As equiangular tube pitching was used, the plate was divided into hexagonal cylinders surrounding each tube.

[48] This chapter is based on Reference 7. For details of the mathematical derivation the reader is referred to References 6 and 8.

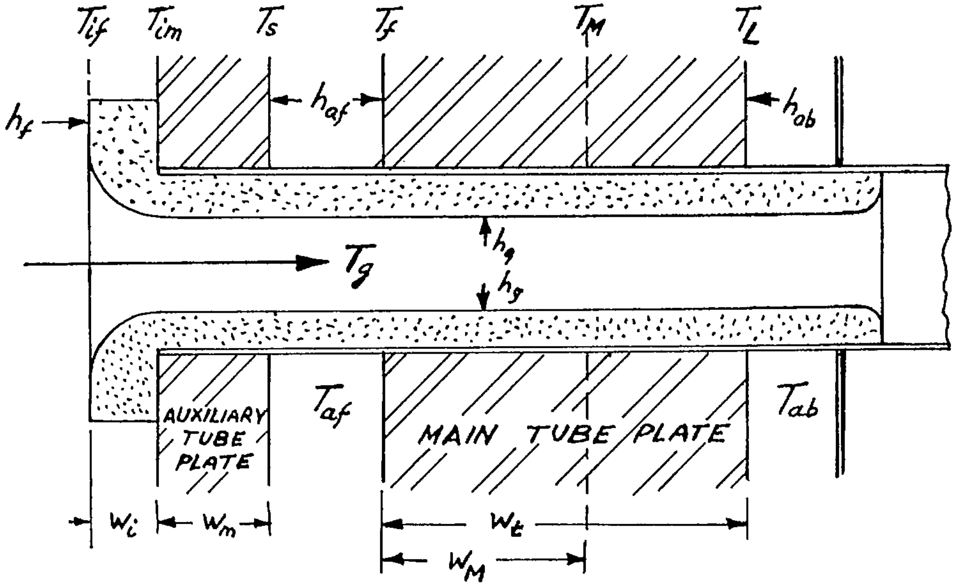

Figure A2-1. Hot-tube-plate arrangement—definition of symbols.

The outside diameter of a circular cylinder having the same cross-sectional area as the hexagonal cylinder was calculated. The circular cylinder was then considered as the only region of the tube plate affected by its tube. This assumption imposed the boundary condition that $\frac{\delta T}{\delta r} = 0$ for the outer boundary of the circular cylinder.

The assumptions involved in the method are:

(a) The front surface, directly exposed to the impingement of the hot gases, assumes their total temperature. Essentially, this is the assumption that the front-surface heat-transfer coefficient is infinite.

(b) Heat-transfer coefficients as given by McAdams (9) for cross-flow normal to a bank of tubes are applicable for estimating the heat-transfer coefficients for the cooling fluid.

(c) Decrease in temperature of the hot gases in the entrance region is ignored.

(d) Average values of the empirical coefficients determined in the Gas Dynamics Laboratory experiments for heat transfer in the entrance region of tubes with metal liners are used.

(e) Temperature distribution in the hot gases approaching the plate is assumed to be uniform.

(f) Cooling-fluid heat-transfer coefficients are independent of the radius of the tube plate. Calculations with the previous assumptions substantiate this.

(g) These last two assumptions imply that there will be no systematic variation of tube-sheet temperature with the radius. There will, of course, be localized radial temperature gradients around each tube, but these will be essentially the same at all tube-sheet radii.

Owing to the difficulty of finding an exact solution of the basic steady-state temperature equation for the given boundary conditions, an approximate method was employed. This method consisted in ignoring radial heat flow in the metal surrounding each tube but allowing for radial heat flow into the metal. Corrections were effected by assuming 1) that the same amount of heat flowed radially in the metal as was considered previously to have flowed into the metal from the tube, and 2) that the radial heat flow continued to the outer radius of the area affected by the tube. Both of these assumptions are conservative, since the actual heat flow in the radial direction will be reduced by the resistance of the metal, and all the heat will not flow to the outer radius. The calculated radial gradient will hence be greater than the actual gradient, with the result that the maximum temperatures will be higher than the actual temperatures.

The derivation of the differential equation and its solution for the existing boundary conditions would be too lengthy to reproduce here, and the reader is referred to Reference 8 for the details of the analysis.

The following equations were derived:

$$\frac{T_g - T_{af}}{T_g - T_{im}} = \frac{k_i \phi_i}{k_m \phi_m} \left[\frac{\dfrac{k_i \phi_i}{h_f} e^{\phi_i W_i} - \cosh \phi_i W_i \left(\dfrac{k_i \phi_i}{h_f} - 1 \right)}{\dfrac{k_i \phi_i}{h_f} e^{\phi_i W_i} - \sinh \phi_i W_i \left(\dfrac{k_i \phi_i}{h_f} - 1 \right)} \right] \times$$

$$\times \left[\frac{k_m \phi_m}{h_{af}} \cosh \phi_m W_m + \sinh \phi_m W_m \right] +$$

$$+ \frac{k_m \phi_m}{h_{af}} \sinh \phi_m W_m + \cosh \phi_m W_m \qquad [1]$$

$$\frac{T_g - T_s}{T_g - T_{im}} = e^{\phi_m W_m} - \sinh \phi_m W_m \left[\frac{\dfrac{T_g - T_{af}}{T_{im} - T_g} + e^{\phi_m W_m} \left(\dfrac{k_m \phi_m}{h_{af}} + 1 \right)}{\dfrac{k_m \phi_m}{h_{af}} \cosh \phi_m W_m + \sinh \phi_m W_m} \right] \qquad [2]$$

$$\frac{e^{2\phi_t W_M} (1 + \Psi_{af}) + (1 - \Psi_{af})}{e^{2\phi_t W_M} (1 + \Psi_{af}) - (1 - \Psi_{af})} = \frac{1}{e^{2\phi_t W_M} (T_g + \Psi_{af} T_{af}) - (T_g - \Psi_{af} T_{af})} \times$$

$$\times \left[\frac{4 (T_{ab} - T_g) e^{2\phi_t W_M}}{(1 + 1/\Psi_{ab}) e^{\phi_t W_t} + (1 - 1/\Psi_{ab}) e^{-\phi_t W_t} e^{2\phi_t W_M}} + \right.$$

$$\left. + e^{2\phi_t W_M} (T_g + \Psi_{af} T_{af}) + (T_g - \Psi_{af} T_{af}) \right] \qquad [3]$$

$$\frac{T_{ab} - T_g}{T_M - T_g} = \frac{1}{2} \left[\left(1 + \frac{1}{\Psi_{ab}} \right) \frac{e^{\phi_t W_t}}{e^{\phi_t W_M}} + \left(1 - \frac{1}{\Psi_{ab}} \right) \frac{e^{-\phi_t W_t}}{e^{-\phi_t W_M}} \right] \qquad [4]$$

$$T_f = \frac{e^{\phi_t W_M} (T_g + \Psi_{af} T_{af}) - e^{-\phi_t W_M} (T_g - \Psi_{af} T_{af})}{e^{\phi_t W_M} (1 + \Psi_{af}) - e^{-\phi_t W_M} (1 - \Psi_{af})} \qquad [5]$$

$$\text{or} \quad T_f = \frac{2(T_M - T_g) + e^{\phi_t W_M} (T_g + \Psi_{af} T_{af}) + e^{-\phi_t W_M} (T_g - \Psi_{af} T_{af})}{e^{\phi_t W_M} (1 + \Psi_{af}) + e^{-\phi_t W_M} (1 - \Psi_{af})} \qquad [6]$$

$$T_L = \frac{(T_g - T_M)}{2\Psi_{ab}} \left[\frac{e^{\phi_t W_t}}{e^{\phi_t W_M}} - \frac{e^{-\phi_t W_t}}{e^{-\phi_t W_M}}\right] + T_{ab} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [7]$$

Correction for radial gradient:

$$\frac{T_2 - T_1}{T_g - T_1} = -\frac{U_g\, d_1}{2k_t} \ln \frac{d_2}{d_1}$$

where,

T_g = gas temperature, °R
T_{af} = front-passage cooling-fluid temperature, °R
T_{ab} = rear-passage cooling-fluid temperature, °R
T_{im} = temperature at boundary between insulation and metal, °R
T_s = auxiliary-tube-sheet rear-surface temperature, °R
T_f = main-tube-sheet front-surface temperature, °R
T_M = main-tube-sheet maximum temperature, °R
T_L = main-tube-sheet rear-surface temperature, °R
k_i = conductivity of insulation, Btu/ft^2 °F sec/in.
k_m = conductivity of auxiliary-tube-sheet material, Btu/ft^2 °F sec/in.
k_t = conductivity of main-tube-sheet material, Btu/ft^2 °F sec
h_f = heat-transfer coefficient, front surface, Btu/ft^2 °F sec (assumed infinite)
h_{af} = heat-transfer coefficient, front cooling passage, Btu/ft$_2$ °F sec
h_{ab} = heat-transfer coefficient, back cooling passage Btu/ft^2 °F sec
w_i = thickness of insulation, inches
w_m = thickness of auxiliary tube-sheet, inches
w_M = distance from front surface of main tube-sheet to point of maximum temperature, inches
w_t = thickness of main tube-sheet, inches

$$\phi_i = \sqrt{\frac{U_{gi}\,\pi d_i}{k_i\, A_w}}$$

$$\phi_m = \sqrt{\frac{U_{gm}\,\pi d_i}{k_m\, A_w}}$$

$$\phi_t = \sqrt{\frac{U_{gt}\,\pi d_i}{k_t\, A_w}}$$

$$\Psi_{af} = \frac{h_{af}}{k_t \phi_t}$$

$$\Psi_{ab} = \frac{h_{ab}}{k_t \phi_t}$$

U_g = average tube-entry-region heat-transfer coefficient for area considered, Btu/ft^2 °F sec (evaluated at tube ID—0.93″)
d_1 = inner diameter of region associated with each tube, in inches
d_2 = outer diameter of region associated with each tube, in inches
T_1 = Temperature at diameter d_1, °R
T_2 = Temperature at diameter d_2, °R
A_w = cross-section of region associated with each tube, in inches^2

Accepted values for conductivity were used and allowance was made for their variation with temperature. Heat-transfer coefficients were calculated by accepted formulae and from the results of tests conducted in the Gas Dynamics Laboratory.

These tests pertained to the coefficients in the tube entrance region, where heat-transfer rates are considerably higher than along the remainder of the tubes. A test rig was constructed which made possible the calculation of local coefficients at small increments of the tube length near the entry (6,30), and methods of reducing the high heat-transfer rates by means of tube inserts were investigated.

Figure A2-2 gives representative results. The outcome of these and other tests was that ceramic inserts were specified for the hot heat exchanger.

The coefficients obtained in these tests were used in the temperature equations, and the final temperature distributions obtained are shown in Figure 13 on page 18.

These temperature distributions (simplified to straight-line variations) were used to determine the thermal stresses in the tube-sheet.

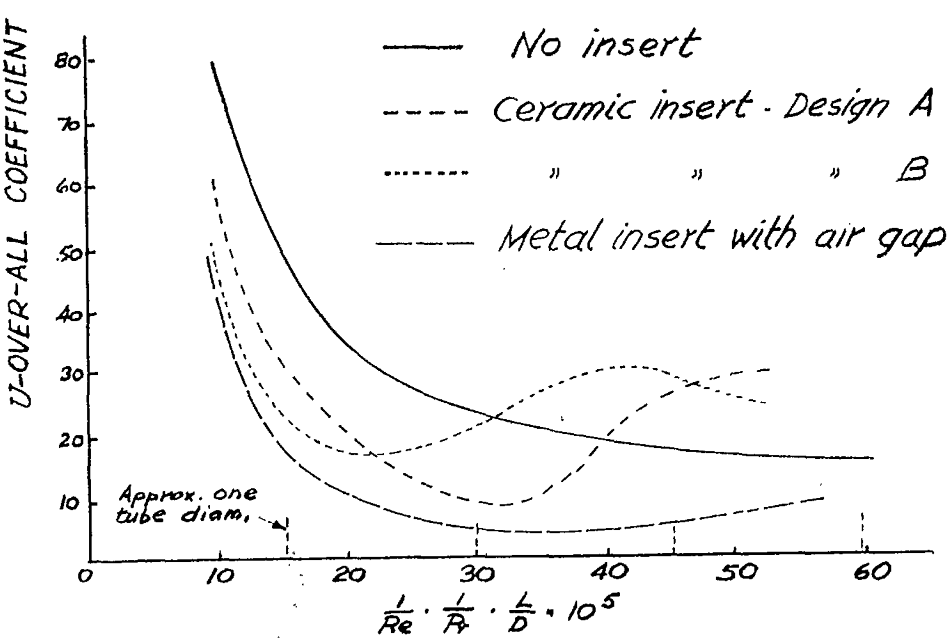

Figure A2-2. Effect of various inserts on entrance length coefficients.

Stress Analysis

The tube plate consists of an annular tubed region surrounding a solid central portion. The design conditions were as follows:

Differential pressure across plate	48.1 psi
Outer diameter	37 inches
Inner diameter of tubed region	15 inches
Estimated maximum temperatures (cooling-fluid flow—0.05 lb/sec)	
Front surface	1,203 °F
Maximum (1.06″ from the front surface)	1,319 °F
Back surface	1,271 °F
Number of tubes	504
Tube OD	1 inch
Ratio of pitch to diameter of tubes	1.25
Pitching	equiangular

To allow for the stiffening effect of the tubes, the actual differential pressure in the tubed region was reduced by an amount equal to the fictitious pressure resulting if the total axial tube tension were assumed to act uniformly over the whole tubed area. The reduced pressure in this region was taken as 23.75 psi. Allowance was made for the effect of the tube holes in the plate by a method devised by I. Malkin (31). This method consists in evaluating elastic constants (modulus of elasticity and Poisson's ratio) for a solid tube plate of the same dimensions composed of an imaginary material which would behave in the same manner as the actual perforated sheet. These constants are then inserted in the ordinary plate formulae so that the stresses and deflections can be computed. Stress formulae for a circular plate simply supported at the edge were employed, as this would result in calculated stresses higher than those which would actually exist if there were any edge fixity. This imparts an implicit factor of safety to the calculations. Starting from the basic formulae for stress in a flat circular plate in terms of the deflection angle, expressions for the circumferential and radial stresses were derived. The stress equations are:

$$p_z = \frac{Em}{m^2-1}\, y \left[m\,\frac{\theta}{x} + \frac{d\theta}{dx} \right]$$

$$p_x = \frac{Em}{m^2-1}\, y \left[\frac{\theta}{x} + m\,\frac{d\theta}{dx} \right]$$

where

$$\frac{\theta_1}{x} = A_1 + \frac{B_1}{x^2} - \frac{1}{4}\, k_1\, \frac{x^2}{R_1^2}$$

$$\frac{d\theta_1}{dx} = A_1 - \frac{B_1}{x^2} - \frac{3}{4}\, k_1\, \frac{x^2}{R_1^2}$$

$$\frac{\theta_2}{x} = A_2 + \frac{B_2}{x^2} - \frac{1}{4}\, k_2\, \frac{x^2}{R_2^2} + C_2 \left(\frac{1}{2} - \ln x \right)$$

$$\frac{d\theta_2}{dx} = A_2 - \frac{B_2}{x^2} - \frac{3}{4}\, k_2\, \frac{x^2}{R_2^2} - C_2 \left(\frac{1}{2} + \ln x \right)$$

E = modulus of elasticity, psi
m = inverse of Poisson's ratio
y = distance in axial direction from neutral plane, inches
θ = deflection angle of plate
p_z = circumferential stress, psi
p_x = radial stress, psi
R_1 = radius of central region, inches
R_2 = outer radius, inches
x = radial distance from centre, inches
Subscript 1 refers to the central region.
Subscript 2 refers to the outer annular region.

$$k_1 = \frac{3(m^2_1 - 1)\ p\ R_1^2}{E_1\ m_1^2\ t^3}$$

$$k_2 = \frac{3(m^2_2 - 1)p_r\ P_2^2}{E_2\ m_2^2\ t^3}$$

$$C_2 = \frac{3(m_2^2 - 1)\ (p - p_r)\ R_1^2}{E_2\ m_2^2\ t^3}$$

t = plate thickness, inches
p = actual pressure on plate, psi
p_r = adjusted pressure, outer region, psi

A_1, B_1, A_2, B_2, are arbitrary constants which may be determined from the following boundary conditions:

(1) When $x = 0$, $\theta = 0$
(2) When $x = R_1$, $p_{x1} = p_{x2}$
(3) When $x = R_1$, $p_{z1} = p_{z2}$
(4) When $x = R_2$, $p_{x2} = 0$

When these boundary conditions were applied the values of the arbitrary constants were found to be:

$$A_1 = \frac{E_2 m_2(m_1 - 1)}{E_1 m_1(m_2^2 - 1)}\left\{\left(1 - \frac{R^2_1}{R_2^2}\right)\left[\frac{m^2 - 1}{2}\left(\frac{k_1}{2}\frac{R_1^2}{R_2^2} + C_2 - \frac{E_1 m_1 k_1}{2E_2 m_2}\right)\right.\right.$$
$$\left.\left. + \frac{k_2}{4}(1 + 3m_2)\right] + C_2(1 + m_2)\ln\frac{R_2}{R_1}\right\} + \frac{k_1}{4}\frac{(1 + 3m_1)}{(1 + m_1)}$$

$$B_1 = 0$$

$$A_2 = \frac{R_1^2(1 - m_2)}{2R_2^2(1 + m_2)}\left(\frac{k^2}{2}\frac{R_1^2}{R_2^2} + C_2\right) - \frac{E_1 m_1(1 - m_2)\ k_1}{4\ E_2 m_2(1 + m_2)}\frac{R_1^2}{R_2^2}$$
$$+ \frac{k_2}{4}\frac{(1 + 3m_2)}{(1 + m_2)} - C_2\left[\frac{(1 - m_2)}{2(1 + m_2)} - \ln R_2\right]$$

$$B_2 = \frac{E_1\ m_1}{4\ E_2 m_2} R_1^2\ k_1 - \frac{R_1^2}{2}\left(\frac{k_2}{2}\frac{R_1^2}{R_2^2} + C_2\right)$$

From preliminary calculations of the maximum stress in the tube plate, it was determined that a thickness of from 1.50 inches to 2.00 inches would be satisfactory. Finally, a complete calculation indicated that a 1.75-inch-thick plate of Type 310 steel would sustain the load at the estimated temperatures without exceeding the desired creep rate of 1 per cent in 10,000 hours. The calculated stresses are shown in Figure A2-3. It will be noted that the maximum stress of 4,770 psi occurs at the center of the plate, which, being relatively remote

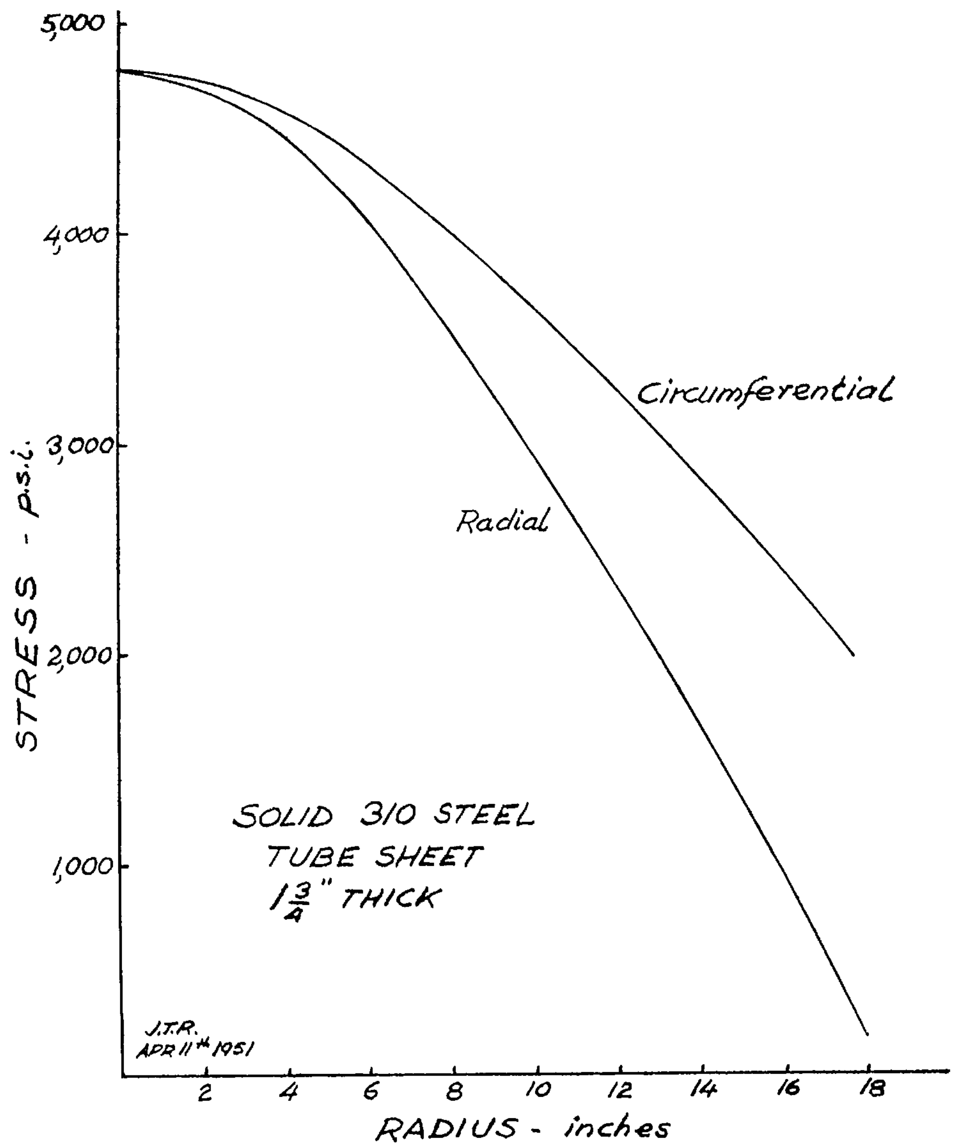

Figure A2-3. Tube-sheet stresses.

from the tubes and well insulated from the hot gases, should be at a considerably lower temperature than the maximum plate temperature. At the inner edge of the tubed region, the maximum stress is 4,100 psi.

Thermal stresses were calculated according to the method of Timoshenko (11). The maximum thermal stress in a flat plate due to a linear temperature gradient ΔT across the plate is:

$$p_t = \frac{\alpha \;\; E \;\; \Delta T}{2\left(1 - \frac{1}{m}\right)}$$

where α = linear coefficient of expansion, in./in. °F.
Other symbols are as already given.

Since the temperature gradient in the plate was not linear, it was necessary to adjust this formula accordingly. This was accomplished by calculating the temperature difference between the neutral surface and each face of the plate. These values were then employed in the foregoing formula to estimate the stress at each face separately.

The equivalent maximum stress due to the effect of the combined stresses in the different planes was calculated by the maximum-shear-strain energy theory as recommended for combined stresses under creep conditions (12). At the edge of the tubed region the maximum stress was 7,840 psi.

Appendix 3

FURNACE TESTS[49]

The original design of the furnace is shown diagrammatically in Figure A3-1. The furnace consisted of a 24-inch-diameter cyclone combustor and a mixing section.

The cyclone was lined with plastic chrome-ore, applied on studs $\frac{3}{8}$ inch in diameter by $\frac{3}{4}$ inch long on a 1-inch-square pitching. The mixing section was a 3-foot-square section approximately 6 feet long and contained three baffles. The walls consisted of 5 inches of refractory brick plus $2\frac{1}{2}$ inches of insulating brick.

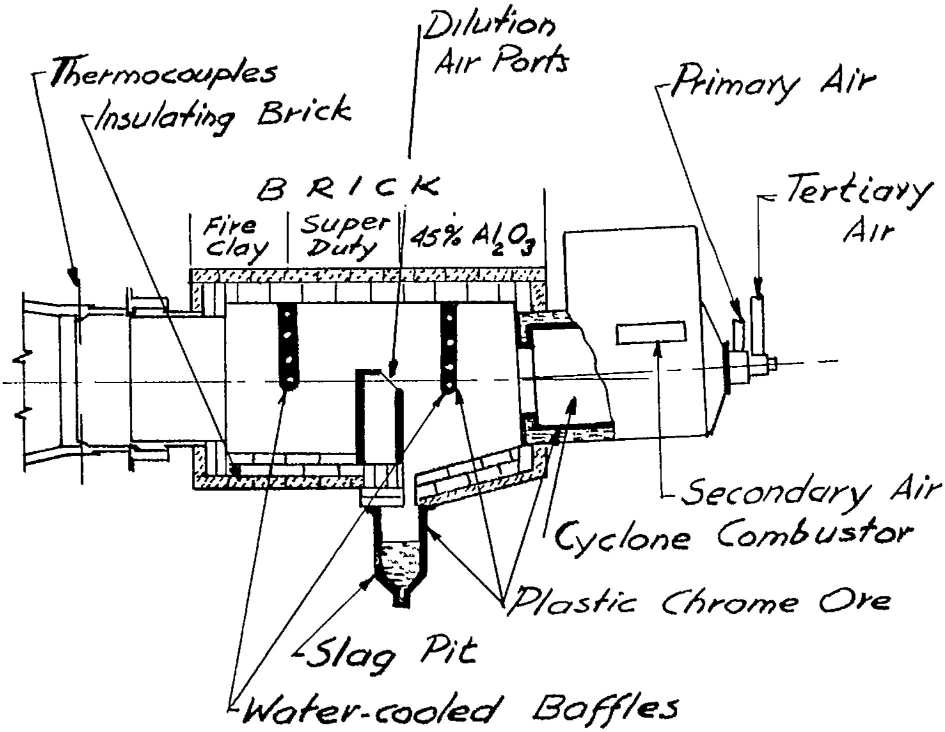

Figure A3-1. Diagram of furnace.

The purpose of the mixing section was to remove the slag, which was carried over from the cyclone or combustion zone of the furnace, and to reduce the temperature of the gas by diluting it with a proportion of the turbine exhaust air. The first baffle forced the hot gas and slag down, depositing the slag on the floor of the furnace. The gas then passed over the second baffle, where the dilution air was injected. The third baffle afforded additional mixing and reduced the radiation to the furnace outlet.

[49] Based on Reference 13.

The slag pit was bolted to the bottom plate of the furnace with a 4-inch-square hole at the top for the slag to enter. The slag entering the pit was kept molten by a breather, which maintained a flow of hot gas through the entry. The breather was a 2-inch pipe connecting the slag pit to the furnace in such a way as to make it possible to use the pressure drop of the dilution baffle to induce a flow of hot gas. The pit was operated half full of water with continuous supply and an overflow system, and a rotating screw was attached to the bottom of the slag pit to extract the slag in granular form after it was quenched in the water.

TEST ARRANGEMENT

Figure A3-2 is a diagram of the test arrangement for the furnace. The furnace was supplied with the gas exhausted from a gas turbine, in a manner identical with that of the full coal-burning turbine. The gas turbine obtained its heat from the combustion of kerosene in the conventional combustors.

The heat produced in the furnace was quenched by water sprays before discharging to the exhaust stack.

TESTS

The furnace was first run on December 10, 1952, and after some preliminary adjustments were made so that the oil for preheating would be used to the best advantage, coal was burnt for the first time on December 23. In the next six weeks many tests were made to try the effects of various modifications and improvements. By the middle of February 1953, a reasonable combustion performance appeared attainable.

Some six weeks were then spent in detailed calibrations and checking, so that a complete record of the performance of the furnace would be available.

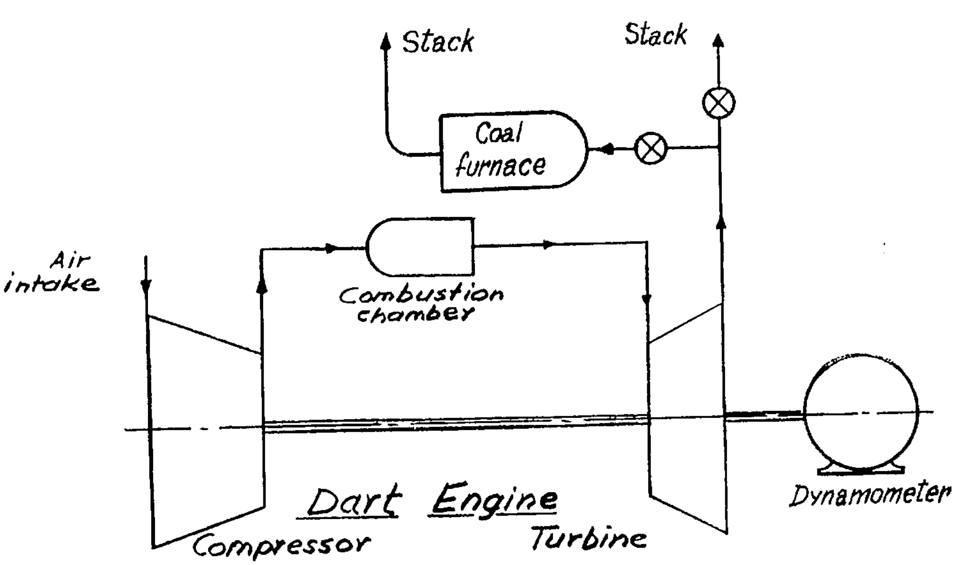

Figure A3-2. Diagram of test arrangement.

The results of these tests were used in determining the suitability of the furnace for operating the engine on its preliminary tests. The test program was terminated on March 27, when a minor failure occurred in the turbine engine being used as a slave. As the test schedule was nearly completed, it was decided to curtail the tests at this point and carry on with the building of the complete plant.

TEST DATA AND RESULTS

Analysis of the results was not a simple matter, because some of the oxygen was previously burnt out of the combustion air by the kerosene used as fuel in the slave turbine that supplied the preheat air. Furthermore, the different streams of air for primary, secondary, tertiary and dilution purposes had different heat losses and so arrived at the furnace at different temperatures.

The heat supplied by the coal was accounted for by the following:

(a) the increase of sensible heat, i.e., the temperature of the gases. (This represents what we would regard as the useful output of the furnace).
(b) the carryover of unburnt fuel
(c) losses due to incomplete combustion
(d) losses to the cooling water around the primary furnace and the baffles
(e) losses to atmosphere from the mixing chamber and ducting.

Items (a), (c) and (d) can be measured more or less exactly by thermocouple traverse, analysis of a representative sample of exhaust gas, and heat balance on the cooling water. The heat losses to atmosphere, which were considerable in

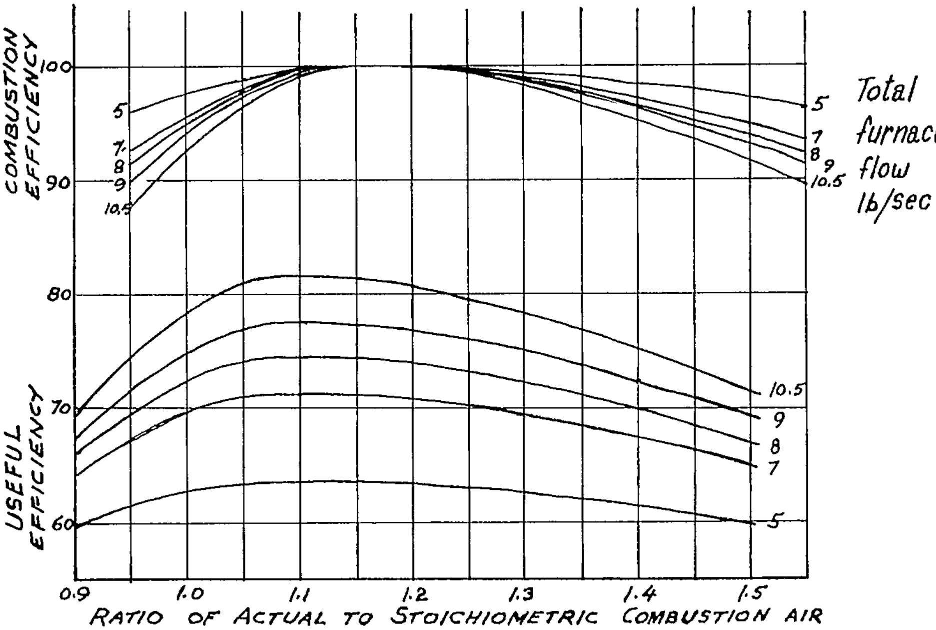

Figure A3-3. Furnace efficiencies.

this installation, were found by operating the furnace on kerosene. With kerosene combustion, items (b) and (c) were assumed negligible, and Orsat analysis showed practically no carbon monoxide. Consequently, the heat losses under item (e) could be determined from the heat balance.

It was found that, at the same furnace-outlet temperatures, the heat losses to the water were slightly greater (about 20 per cent) with coal operation than with oil. As the coefficient of heat transfer to the water was high and there was relatively little protective insulation, it would appear that the higher radiation effects obtained with the luminous coal flame increased the over-all transfer coefficient slightly.

On the other hand, the heat losses to the air were limited by more adequate insulation and by a relatively low final coefficient of radiation and convection to atmosphere. Since the increased radiation with coal would therefore have a much smaller effect on the over-all coefficient, it was assumed that losses to atmosphere from the mixing chamber were the same when coal was burned as when oil was used. The validity of this assumption was to some extent confirmed by the final results. By working back from this assumption the complete heat balance could be obtained, and under the best operating conditions, it accounted for practically all the heat supplied in the fuel.

Figure A3-3 shows the combustion and useful efficiencies as functions of the excess-air coefficient for different total mass flows. The combustion efficiency is defined as the heat liberated compared with the heat available in the coal; the useful efficiency is defined as the increase of sensible heat in the furnace gases compared with the heat available in the coal.

The excess-air ratio is simply the actual air flow divided by the air flow which would be chemically correct for combustion, with due allowance for the diminution caused in the oxygen by the previous kerosene combustion. It will

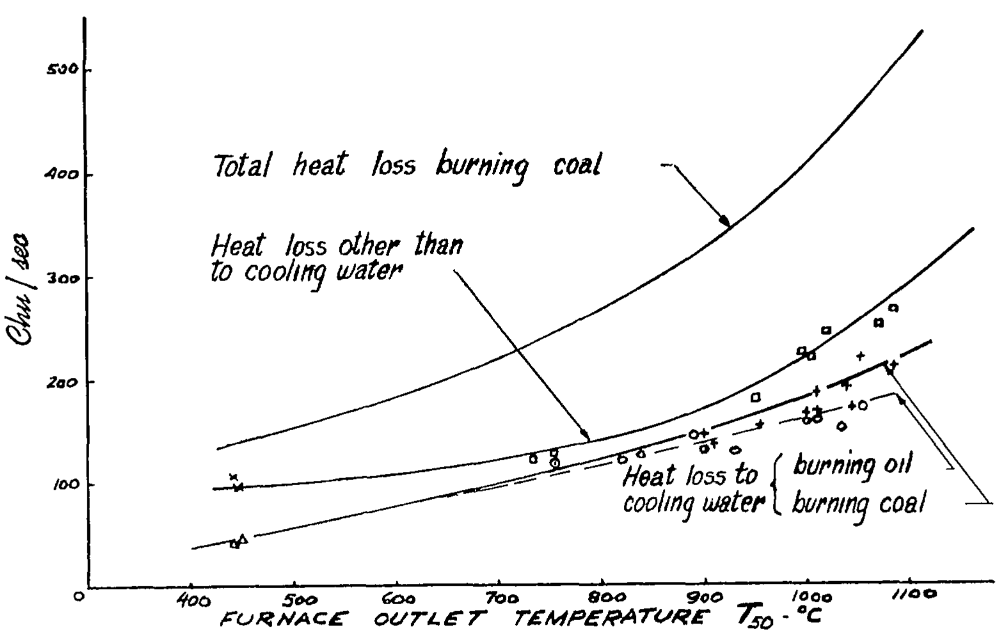

Figure A3-4. Furnace heat losses.

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be seen that between the values of 1.05 and 1.20 for the excess-air coefficient, the combustion efficiency is between 99 to 100 per cent. The statement that the combustion efficiency is 100 per cent really means that it is 100 per cent of the value obtainable by burning oil, which was not directly measured but was assumed to be 100 per cent on the basis of the Orsat analysis.

The useful efficiency, which is the prime consideration here, reaches a peak of about 81 per cent at the highest flow but falls off rapidly as the total mass flow decreases. The reason for this, of course, is that the heat losses do not depend upon the actual rate of air flow but upon the outlet temperature, and therefore become a relatively greater fraction of the total heat release as the flow decreases.

Figure A3-4 shows the heat losses and the difference between the heat lost to cooling water on oil and that lost on coal. The heat loss other than to cooling water is the curve of losses derived from the calibration tests on oil, and it is assumed to be the same for coal.

ASH

Throughout the series of tests, some 16,200 pounds of coal were burnt, and the ash content of the coal as fired was 7 per cent. Of the ash that this produced, some 64 per cent was collected in the slag pit, 17.5 per cent was collected as fly ash in the outlet duct, and 8.75 per cent was found in the drain trench, which was really an overflow from the outlet duct.

In all, therefore, 90 per cent of the ash was apparently collected. However, some of the slag-pit content, which analysis showed to contain virtually no combustible, was undoubtedly due to fused bricks or lining which had run down into the slag pit. The fly-ash samples showed, in some cases, only 24 to 40 per cent of ash, the remaining being combustible matter.

As the ash balance could be obtained only over a long period of running, these results include not only the tests in which the furnace operated under the correct conditions, but also the preliminary tests in which the conditions were far from correct. It was found that under operation with an excess-air coefficient of 1.1, the carryover was very small, and consideration of the over-all heat balance shows that its combustible content must have been negligible.

As soon, however, as the excess-air coefficient was reduced, carryover increased owing to an insufficiency of oxygen. If the coefficient was increased to values much greater than 1.1, the time available for combustion was lessened and the temperature was lowered and there was again an increase in the carryover of unburnt combustible.

FURNACE CONDITIONS

After the tests the furnace was fully inspected. The lining of the primary chamber of the cyclone was generally in good condition, having a slag coating whose variations between 0.25 inch and 1.6 inch in thickness depended on the circumferential distribution of the air and also, apparently, on the actual distribution of the temperature within the furnace. There was considerable erosion ($\frac{3}{4}$ inch) of the fire-brick in the hottest part of the mixing zone. The bricks used there were 45 per cent Al_2O_3 fire-brick, Cone No. 34.

In one corner, where its edge was exposed, the brick was eroded to its full depth. Some bricks containing 62 per cent Al_2O_3 were left in the hottest part of the furnace for test purposes and appeared to stand up well. The mortar was eroded more than the bricks, but chrome mortars and castables seemed to stand up better. The many cracks that appeared in the brickwork are attributed to the thermal shock resulting from frequent quick (20-minute) heating and, in some cases, to insufficient provision for thermal expansion.

DISCUSSION

In general, the furnace behaved well. Complete remote control was easy, and the ignition and preheating systems, after due development, were satisfactory. The response of furnace-exhaust temperature to changes in the rate of coal-feeding was rapid, and a rate of 180°F per minute was often recorded.

The main cause for criticism of the system as a whole is the excessive heat losses from the mixing chamber. In building the furnace for the engine tests a new design was evolved which uses dilution air to return most of the heat transmitted through the brickwork back into the system.

The coal handling and feeding systems worked well and permitted accurate measurement and rapid changes of coal rate. The actual feeding system comprised a vibrating feeder and a compressed-air injector. The vibrating feeder used as the main control during the tests gave minor trouble from time to time. Rig tests of the injector suggest that it is capable of feeding and controlling at least as well, if not better, than the vibrating feeder and that it would operate more reliably.

The automatic slag-removal screw, which gave continual trouble, was finally replaced with a box big enough to recover all the slag produced in one run.

CONCLUSIONS

These preliminary tests showed that the turbine would be suitable for first trials of the complete exhaust-heated gas turbine. Performance was excellent from the viewpoint of combustion, and a large part of the ash was recovered as slag. The heat losses in the mixing section were too high, giving a useful efficiency range of 60 to 80 per cent, and the refractories in the hottest part of the mixing section did not stand up under service conditions. It was proposed to overcome these difficulties by passing cooling air over the back of the refractory brick and using better-quality brick.

Appendix 4

SPECIFICATIONS AND DETAILS OF EQUIPMENT

(1) *Starting engine*

Dodge T-120—Six cylinder commercial truck engine fitted with a normal single-plate clutch and a five-speed transmission.

(2) *Dynamometer*

D.P. x R6 Heenan and Froude water dynamometer coupled to the turbine by a modified Dominion Gearflex coupling.

(3) *Dart gas turbine*

From November 1953 to March 30, 1954—Rolls Royce Dart 8 turbo-propellor engine with a reduction ratio of 9.4:1.
From May 19, 1954, to October 16, 1956—Rolls Royce Dart No. 16; reduction ratio 11:1.
From October 31, 1956, to December 1956—Rolls Royce Dart No. 13; reduction ratio 11:1.

(4) *Air intake*

Square bell-mouth nozzle with 109-square-inch throat area (10.455″ × 10.475″); discharge coefficient 0.99 (as obtained by calibration).

(5) *Compressor-outlet manifold*

Seven-branch carbon-steel rectangular manifold connected to the seven compressor outlets by rubber sleeves and ducted to the shell-side inlet of the cold heat exchanger.

(6) *Cold heat exchanger*

36-inch-diameter shell; 760 one-inch OD × 20-gauge (.035″).
13-foot tubes on $1\frac{1}{4}$-inch triangular pitch.
Total surface 2,600 square feet.
Packed floating-head construction.
Material: all carbon steel.

(7) *Hot heat exchanger*

From November 1953 to April 1955—Shell and tube, fixed tube-sheet exchanger, expansion joint in shell, 38-inch OD, $\frac{1}{2}$-inch-thick carbon-steel shell fitted with a stainless-steel liner inside and equipped for cooling.
498 one-inch, 20-gauge Nimonic 75 tubes $18\frac{1}{2}$ feet long, on $1\frac{1}{4}$-inch triangular pitch, total surface 2,300 square feet.
15-inch OD internal transfer duct; 3-foot-long parallel flow section at hot end.
January to November 1956—As above, except for 500 tubes and counterflow throughout.

(*8*) *Turbine-inlet ducting and manifold*

Three 12-inch-diameter riser ducts connecting to an 8-inch-diameter manifold surrounding the turbine and thence to seven turbine inlets via spherically seated connecting sleeves; all stainless steel.

(*9*) *Bypass duct*

16-inch diameter, containing 8.2-inch-diameter Venturi throat (first series of tests only).

(*10*) *Cyclone combustion chamber*

As described in text and shown in Figure 14 and Figure 37.

Refractories:	cyclone zone, baffles....	plastic chrome ore
	mixing zone...........	silicone carbide
	outlet duct............	fire-brick-lined with Durolite castable refractory.

(*11*) *Exhaust stack*

32 feet high, made of 4-foot-diameter reinforced-concrete pipe 5 inches thick; equipped with water sprays for cooling exhaust gases.

(*12*) *Sound absorption*

Whole test cell lined with sound-proofing tiles.
Air intake in the laboratory loft surrounded by a system of baffles made of sound-proofing material.

(*13*) *Coal-handling Equipment*

1 elevator, two-step portable type.
1 coal-hopper, capacity 5 tons.
2 hopper bases with shutters and feed-screw with 2-hp.-drive motor.
1 hammer-type 'Jeffrey' coal crusher driven by a 10-hp. motor and connected by a vertical pipe to
1 Rotoclone Type-D blower with cyclone separator (driven by a 5-hp. motor).
1 coal-hopper placed on a balance scale for fuel measurement (capacity $2\frac{1}{4}$ tons).
1 additional coal-hopper for storing crushed coal (capacity 1 ton) placed directly above measuring hopper and discharging into it.
1 coal feeder

November 1953 to January 1954—A Jefferies tubular vibrating feeder regulated with a rheostat and discharging into an injector, from which compressed air conveys the coal to the furnace.

After January 1954—Compressed air used for metering and injection of coal.

Appendix 5

INSTRUMENTATION AND CONTROLS[50]

(as installed on November 13, 1953)

Legend

Measuring-element designation

T-12	Thermocouple (or thermometer) number
P-12	Pressure-tap number
Prefix F	Furnace
Prefix HX	Hot heat exchanger
Prefix CX	Cold heat exchanger
Prefix D	Dart engine

Indicating-instrument designation

G	Galvanometer
P	Potentiometer
R	Recorder (followed by code number of measuring element used)
V	Resistance thermometer gauge

Cycle Temperatures

Location	Measuring-element Code Number	Indicating Instrument
Air Temperatures		
(1) Compressor intake	T-11, 13	P
(2) Compressor delivery	T-21 to 23	P
(3) Cold-exchanger inlet	T-26, 27	P
(4) Hot-exchanger inlet	T-28, 29	P, R (T-29)
(5) Turbine inlet	T-30, 31 to 36	G, R (T-35a, 36a)
(6) Turbine exhaust	T-41 to 45	G
(7) Furnace-inlet streams	T-46 to 49	G, R (T-49b)
Gas Temperatures		
(1) Furnace outlet	T-50 to 58	G, R(T-56a)
(2) Hot-exchanger outlet	T-71 to 74	G, R(T-73)
(3) Cold-exchanger inlet	T-76, 77	G, R(T-77)
(4) Cold-exchanger outlet	T-82 to 84	P, R(T-83)
(5) Exhaust stack	T-91	G

[50] This list enumerates only the most important measuring stations. For a full list see Reference 32.

Cycle Pressures

Location	Measuring-element Code Number	Indicating Instrument
Air Pressures		
(1) Compressor intake	VH 11 to 14, P-10	H_2O manometer, P
(2) Compressor delivery	P-21, 22	H_2O manometer
(3) Cold-exchanger inlet	P-26	H_2O manometer
(4) Hot-exchanger inlet	P-28	H_2O manometer
(5) Hot-exchanger transfer duct[51]	P-31	H_2O manometer
(6) Hot-exchanger outlet	P-30	H_2O manometer
(7) Turbine outlet	P-40	H_2O manometer
(8) Furnace-inlet streams	P-41, 46 to 49	H_2O manometer
(9) Bypass stream	P-45	H_2O manometer
Gas Pressures		
(1) Furnace outlet	P-50	
(11) Hot-exchanger outlet	P-70, 73	
(12) Cold-exchanger inlet	P-75	
(13) Cold-exchanger outlet	P-80	

Plant Temperatures

Location	Measuring-element Code Number	Indicating Instrument
Furnace		
(1) Dilution air	FT 1 to 6	G
(2) Cooling water	FT 11 to 13, T-100a	V
(3) Brickwork	FT 21 to 26	G
(4) Shell	FT 27 to 34	G
Hot Heat Exchanger		
(1) Tube wall	HX 1 to 4	G
	HX 13 to 20	G, R(HX-17)
(2) Air at transfer duct	HX 5 to 8	G, R(HX-7)
(3) Hot tube plate	HX 9 to 12	G, R(HX-11)
	HX 21 to 24	G, R(HX-21, 23)
	HX 25 to 28	G, R(HX-25, 27)
	HX 29 to 32	G
(4) Tube-plate cooling air	HX 33 to 35	P
(5) Shell metal	HX-36 to 51	G
(6) Tube-plate cooling water	T-100b	
	HXT-61, 62	
	HXT-64 to 68[52]	
Dart Engine		
(1) Oil	DT-1	
(2) Bearings	DT-2 to 4	

[51] Refers to transfer from counterflow to parallel-flow section.
[52] Added on December 4, 1953.

Appendix 6

CALIBRATION OF THE NO. 8 DART ENGINE AND PERFORMANCE ANALYSIS OF THE EXPERIMENTAL COAL-BURNING GAS TURBINE

The No. 8 Dart engine (an early-development model of the Rolls-Royce Dart, designated RDA 2), was tested in order to determine its performance characteristics.

A short description of the equipment as installed for the calibration tests follows.

The engine was mounted on the mezzanine floor in the test cell described on page 23.

The air intake was the same as that used for subsequent coal-burning tests.

Fuel was burnt in seven can-type combustion chambers, the spark for the ignitors being supplied from a 110-volt line through two transformers.

The fuel, aviation kerosene plus 2% lubricating oil, was controlled by hydraulic throttle controls. The governor was reset to give a maximum engine speed of 13,600 instead of 15,200 rpm.

A 10-gallon container was used for oil supply, and the hot oil was cooled by being circulated through water-cooled pipes.

The engine breather was extended and piped outside the laboratory.

A Tacho generator was adapted to fit the accessory drive coupling.

The starting motor was supplied with five 6-volt car batteries connected in series.

Cooling air for the low-pressure side of the turbine disk was supplied from the laboratory mains.

An electrically actuated damper was installed in the exhaust duct from the turbine to simulate furnace back-pressure. The exhaust gases were then ducted to the stack.

The engine exhaust temperature was measured by four KLG Chromel Alumel thermocouples installed downstream from the engine exhaust cone and connected to read continuously in the control room. A system of water-spray jets cooled the exhaust gases before the entrance to the stack.

A Heenan and Froude type D.P. × R. 6 water dynamometer was used. This dynamometer was very old and vibrated considerably at high speeds. It was replaced by a new one before the furnace tests were proceeded with (Appendix 3).

Figures A6-1, A6-2, and A6-3 summarize graphically the results of the calibration tests.

The performance calculations for the coal-burning plant are now given, together with the list of assumptions made in calculating the power output.

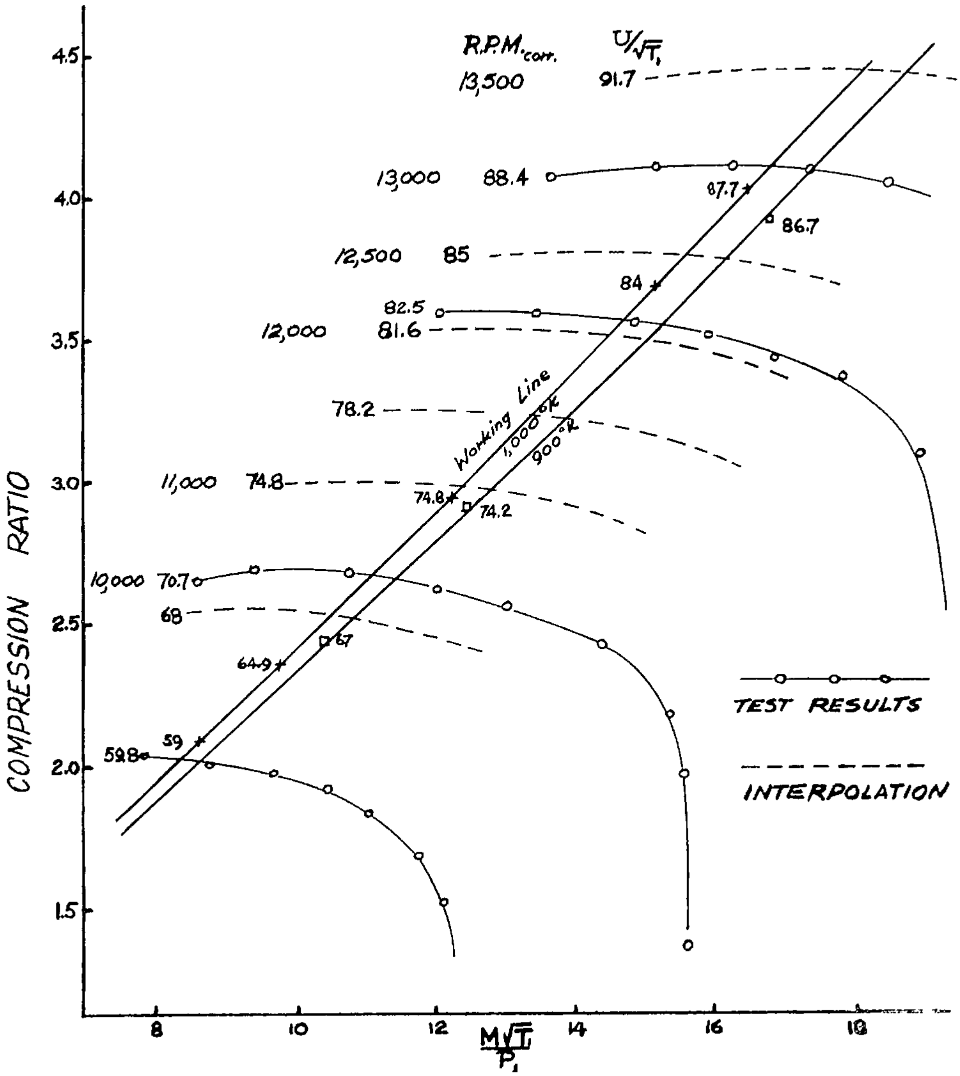

Figure A6-1. Dart-compressor characteristics. Rotor disc diameter: 1st stage 20.0″, 2nd stage 17.2″. Equivalent single-stage diameter $\sqrt{17.2^3 + 20.0^2} = 26.4''$.
Rotor tip speed $U = 0.1152 \times RPM$, ft/sec.

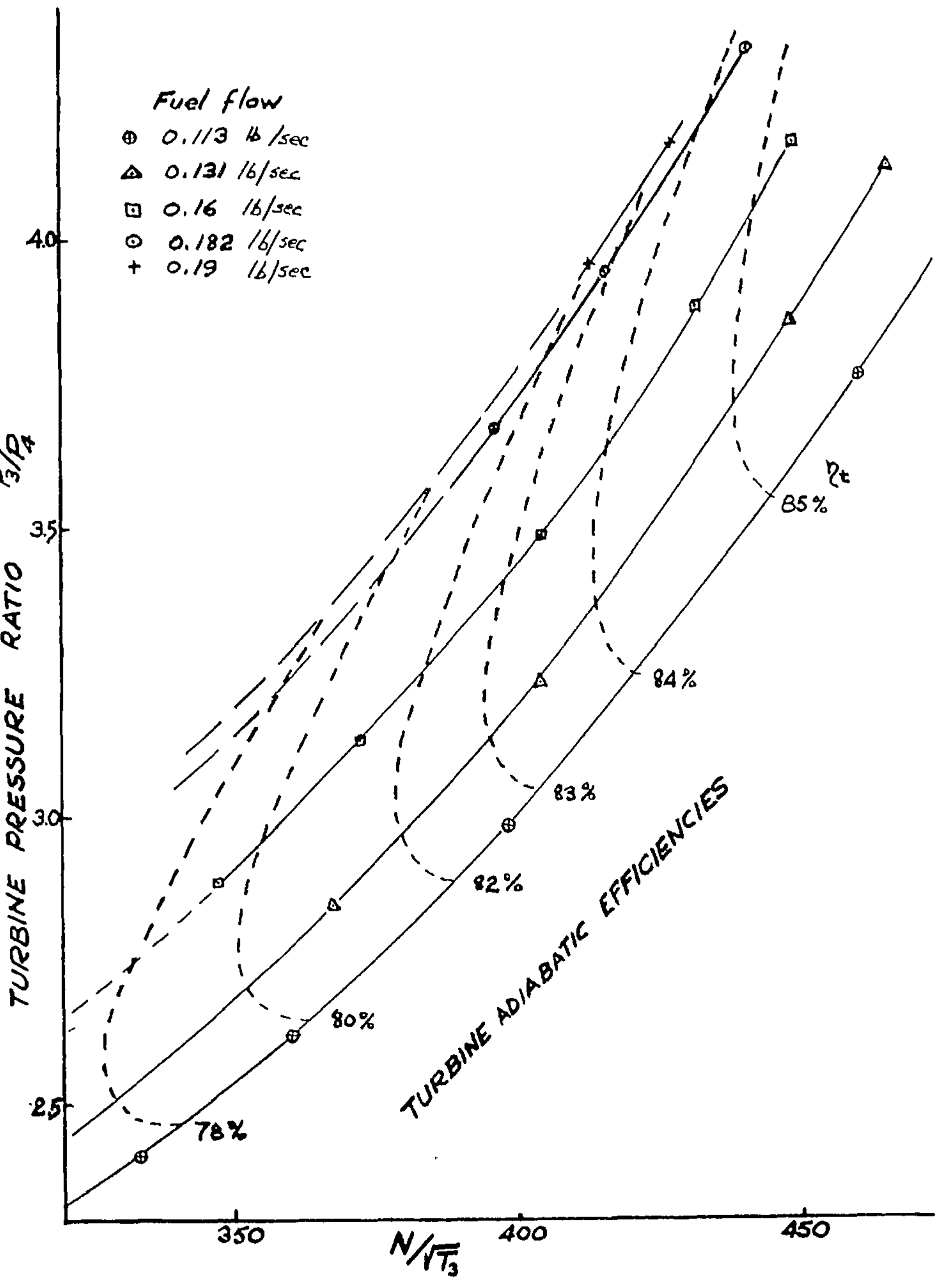

Figure A6-2. Dart-turbine characteristics.

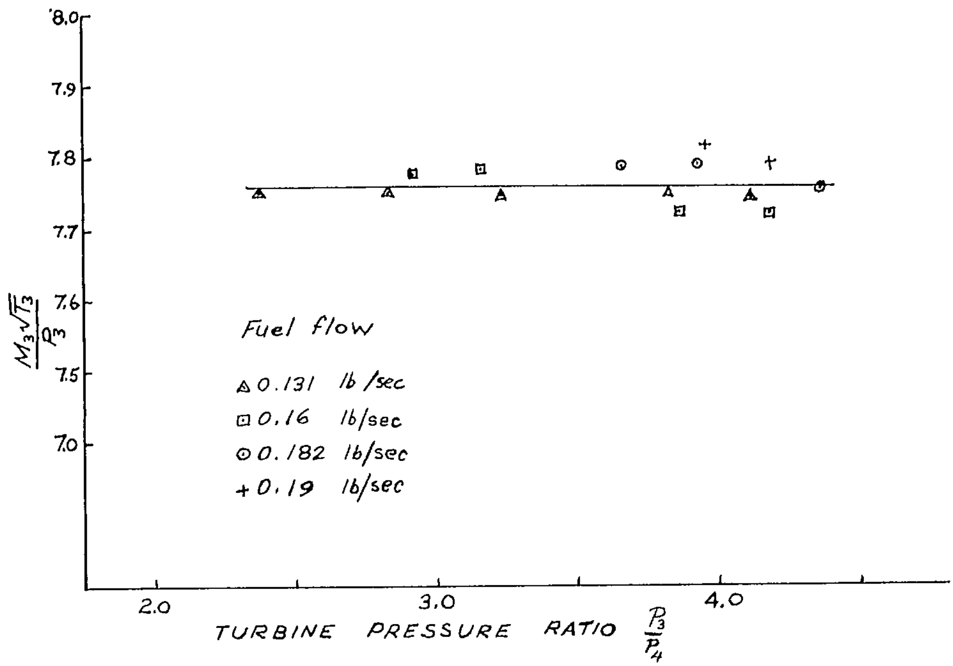

Figure A6-3. Turbine flow vs. pressure ratio.

List of Assumptions Made in the Calculations

Air-side pressure drop P_2-P_3, (% of P_2)	2%
Exhaust pressure drop P_4-P_{atm}, (% of P_{atm})	10%
Cooling air (% of compressor mass flow)	2.5%
Atmospheric pressure P_1, psi	14.7
Temperature of air at compressor entrance T_1,°K	288
Non-dimensional mass flow at turbine inlet $M\sqrt{T_3}/P_3$	7.75
Maximum permissible temperature at the turbine inlet T_3 °K	1,000
Engine speed, rpm	13,000
C_{pa} specific heat of air	from Keenan and
C_{pg} specific heat of combustion gases	Kaye gas tables
Heat losses	none
Combustion losses	none
Compressor temperature rise, °C[53]	$1.18\ N^2 \times 10^{-6}$
% pressure drops proportional to	$\frac{M^2\,T}{P^2}$
Mechanical losses	none

[53] This formula was obtained by correlating actual test results with the calibration curves supplied by the manufacturer.

Performance Calculations

Gas-turbine Components at Maximum Speed and Temperature

Step No.		Operation or Data	
1	$T_1\ ^\circ K$	$= 288\ ^\circ K$	
2	P_1	$= 14.7$ psi	
3	N	$= 13{,}000$ rpm	
4	$\dfrac{M_a\sqrt{T_1}}{P_1}$	$= 16.84$	Selected running point on compressor characteristics (Figure A6-1)
5	$\dfrac{P_2}{P_1}$	$= 4.1$	
6	$T_2 - T_1$	$= 1.18 \times N^2 \times 10^{-6}\ ^\circ C = 199.4$	
7	T_2	$= 487.4$	
8	P_2	$= 60.3$	

From M_a 2.5 per cent is to be subtracted for cooling air.

9 $\quad M_a' = M_a - 0.025\ M_a = 14.23$

The air-side losses in the heat exchanger are assumed to be 2 per cent of P_2.

10 $\quad P_a = 0.02 \times 60.3 = 1.2$

11 $\quad P_3 = P_2 - P_a = 59.1$

12 $\quad \dfrac{M'_a\sqrt{T_3}}{P_3} = 7.75$ from engine tests (Figure A6-3)

13 $\quad T_3 = 1{,}000^\circ K$

Gas-side losses are assumed to be proportional to

$$\left[\frac{M_a'\sqrt{T_4}}{P_4}\right]^2$$

As neither T_4 nor P_4 are known, approximate values of the pressure drop are first assumed and used to obtain a more accurate estimate of the losses. Hence, as first approximation, it is assumed that there is a constant loss of 10 per cent of the exhaust static pressure.

14 $\quad$ Exhaust static pressure $P_8 = P_a = 14.7$ (atmospheric)

15 $\quad$ Assumed gas-side losses $\Delta P_{gas} = 0.1\ P_a$

16 $\quad P_4 = 1.1\ P_a = 16.17$

17 $\quad \dfrac{P_3}{P_4} = \dfrac{59.1}{16.17} = 3.66$

18 $\quad$ Assumed mean specific heat during expansion

$$c_{p34} = 0.265 \text{ Btu/lb } ^\circ F$$

19 $\quad \dfrac{\gamma - 1}{\gamma} = \dfrac{R}{c_p} = \dfrac{96}{1400 \times c_{p34}} = 0.259$

Step No.	Operation or Data
20	$\left[\frac{P_3}{P_4}\right]^{\frac{\gamma-1}{\gamma}} = 1.400$
21	$\left[\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \div \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = 0.2875$
22	$N/\sqrt{T_3} = 411$
23	From Figure A6-2, turbine efficiency $\eta_t = 0.842$
24	$T_3 - T_4 = T_3\ \eta_t \frac{\left[\left(P_3/P_4\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\left(P_3/P_4\right)^{\frac{\gamma-1}{\gamma}}} = 239$
25	$T_4 = 761\ °K$
26	$\frac{M_a\ \sqrt{T_4}}{P_4} = \frac{14.23\ \sqrt{761}}{16.17} = 24.3$
27	Gas-side losses under other conditions

$$\Delta P_{gas\ cond.2} = \Delta P_{gas\ cond.\ 1} \times \frac{(M_a\sqrt{T_4}/P_4)^2{}_{cond.\ 2}}{(M_a\sqrt{T_4}/P_4)^2{}_{cond.\ 1}}$$

The foregoing equation may be used to find the pressure drop under any other conditions.

The specific heat assumed in step 18 can now be checked more accurately.

Step No.	Operation or Data
28	$T_{ave} = \frac{T_3 + T_4}{2} = \frac{1000 + 761}{2} = 880$
29	Revised $C_{p34} = 0.266$
30	$\frac{\gamma-1}{\gamma} = \frac{96}{1400 \times .266} = 0.258$
31	$\left[P_3/P_4\right]^{\frac{\gamma-1}{\gamma}} = 1.398$
32	$\frac{\left[\left(P_3/P_4\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\left(P_3/P_4\right)^{\frac{\gamma-1}{\gamma}}} = 0.285$
33	$T_3 - T_4 = T_3 \times \eta_t \times \text{(Item 32)} = 238$

Step No.	Operation or Data
34	c_{p12} = specific heat of air at average temperature in the compressor = 0.2417
35	Turbine temperature drop required to drive compressor $$\Delta T_{comp} = (T_2 - T_1) \times \frac{c_{p12}}{c_{p34}} \times \frac{1}{0.975} = 186\ ^\circ C$$
36	Turbine temperature drop available for useful power $$\Delta T_{power} = (T_3 - T_4) - \Delta T_{comp} = 52$$
37	$$\text{Useful b.h.p.} = \Delta T_{power} \times M_a \times c_{p34} \times \frac{J}{550} = 501$$

A similar calculation at 11,000 rpm, at which most of the actual testing was carried out, showed a net output of about 200 hp. A more accurate prediction of the hot-exchanger and furnace-pressure losses would have yielded a figure agreeing even more closely with the actual power outputs obtained in tests.

Appendix 7

CHRONOLOGICAL HISTORY OF TESTING PROGRAM

Date of Test	Running Time (hours and minutes)		Description of Test or Modifications	Remarks
	On Oil	On Coal		
			First Series of Tests	
1953				
November 5-9			Motoring runs—calibration of meters.	
November 16			First assisted run—furnace burning oil.	
November 20	5:00		First self-sustained run on oil, 11:52 a.m. Take-off conditions: 6,200 rpm; furnace outlet 1,345°F; turbine inlet 1,057°F.	Hot-exchanger shell overheating—cooling air bypassing cooling passages.
November 27	5:26	2:00	First operational run on coal, 4:08 p.m. Conditions: 11,500 rpm; 1,470°F/1,110°F.	Slag hole blocked. Tube-plate temperature uneven, requiring rearrangement of cooling-water inlets. Cold-exchanger expansion joint seized. Dart 8 vibrating excessively (compressor rotor not balanced).
December 4	1:36	0:16	Attempted five-hour endurance run on coal. Cold-exchanger expansion joint repaired by Vickers; bundle re-assembled properly (was 180 degrees out of turn).	Coal blocked furnace entry. As a result, coal backed into bypass and ignited there, overheating the duct.
1954				
January 22	1:20	6:00	Endurance run—low speed (11,000 rpm).	Turbine-manifold-expansion trouble. Slag drain blocked.
January 29	1:00	5:05	Endurance run—direct coal injection.	HX-shell temperature still high. Manifold-expansion trouble.

Appendix 7—CHRONOLOGICAL HISTORY OF TESTING PROGRAM

Date of Test	Running Time (hours and minutes)		Description of Test or Modifications	Remarks
	On Oil	On Coal		
February 12	3:46	9:26	13-hour non-stop run, most successful to date. Cooling coil applied to shell near gas-outlet nozzles.	Recurrence of manifold-expansion trouble. Considerable improvement in shell temperature of HX. Injection target in cyclone burnt away, with resultant increase in carryover, but combustion good.
March 23	1:53	8:38	Low speed—high-temperature operation. Experimentation with coal injection.	Continued manifold-expansion trouble and slag-hold blocking.
March 30	2:19	0:20	Coal-injection pipe fitted with swirl-vanes. Bypass air control modified to increase sensitivity.	Run interrupted by formation of coke on swirl-vanes and by blockage of primary-air pipe by coal.
May 19-21	15:47	40:03	First run with Dart 16. Second coal-hopper fitted to allow continuous running on coal. Axial coal discharge into cyclone.	Coal injector inefficient at the high coal-discharge rates used in this test. Combustion took place in mixing section; carryover high; slag hole blocked after one hour; could not be opened. Slag 15 inches deep in mixing section at end of run. HX half blocked at entry.
July 5-6	2:01	15:37	Experimentation with coal injection. Plant operated at high temperatures. Inclined coal discharge into cyclone.	Inclined coal feed resulted in better combustion, but feed unsteady. Combustion at HX inlet at times.
August 10-11	10:34	5:34	Coal injector modified.	Combustion improved, but cinder formation observed. Slag hole blocked.
August 12	1:37	—		Slag hole blocked before change made to coal.
August 16	0:44	2:40		Slag hole froze over.
August 18	1:33	14:40	Investigation of effect of T_{50} on HX tube-plate blockage: rpm 11,600°; T_{50} = 1,805°F, T_{30} = 1,250°F (985°C, 678°C).	Stopped owing to slag-box fouling and HX tube-plate blockage.
September 9	2:43	3:10	Hopper-pressurization tests.	Stopped owing to blown manometers.

September 13	1:30	1:17	Hopper-pressure, dilution-air-effect investigation.	Running mostly with assistance or motoring.
September 22	0:42	4:17	Propane-heated-slag-hole test.	No slag flow although drain was heated to 2,200°F.
September 24	2:08	6:39	Continued experimentation with propane-heated slag hole.	Stopped owing to blocked slag hole.
September 30	2:39	0:05	As above.	Slag hole blocked from start of test.
October 13	0:10	3:10	As above.	Center section of slag drain melted away.
October 20	0:47	5:28	New slag drain tried.	Slag drain still not operating satisfactorily.
November 1	1:05	8:03	Four-pipe coal-entry installed. Graphite slag drain used.	Bad cinder formation at coal entry. Slag drain burnt away.
November 8	0:22	0:58	Graphite drain 1.5-inch hole diameter.	Slag hole blocked. Coal entry blocked with cinders.
November 9	0:10	4:24	Graphite drain 2-inch hole diameter.	Slag hole blocked, partly eroded.
November 15	0:13	3:57	Single offset coal entry assembly tried.	Large cinder formations. Slag hole blocking continuously.
November 18	0:12	5:46	Water-cooled slag drain.	Performance better but still not satisfactory. Large cinder formation.
November 24	0:34	7:09	Single-tube fish-tail coal entry. Stainless-steel slag hole without cooling.	Slag hole working fairly well, but gradually eroded by slag running down along drain wall instead of dripping off lip. Local brick erosion in furnace caused overheating.
December 13	3:57	3:23	New starting engine installed. Cyclone relined with Korundal. Re-entrant throat added (10½-inch diameter). Water-cooled coal entry.	Coal-processing-equipment failure. Test stopped when all available coal consumed.
December 15	2:25	21:43	Endurance run, 11,000 rpm, 1,740/1,220°F (950/660°C).	Slag hole open, but icicle formation blocked slag-box screen twice. Otherwise a fairly good run.
December 28	0:07	1:22	Rectangular stainless-steel slag hole with heavy welded S.S. collar and cooling coil.	T_{50} temperatures excessive even at low speeds. 48 tubes found leaking after the test run (owing to corrosion).
1955				
January 25	1:00	6:04	Attempted endurance run.	Coal-processing-gear failure: ran out of coal.

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Appendix 7—CHRONOLOGICAL HISTORY OF TESTING PROGRAM

Date of Test	Running Time (hours and minutes)		Description of Test or Modifications	Remarks
	On Oil	On Coal		
1955—con.				
February 3	4:40	8:25	New water-coal entry with single 2-inch pipe. Central entry with offset delivery. Primary air blanked off. New slag hole.	Coal-feed trouble; excessive coal admitted; overspeeding and overheating. Cinder formation. Six more tubes found leaking after test.
February 28	0:28	6:44	2-inch coal-feed screw in $2\frac{1}{2}$-inch pipe discharging directly into cyclone from auxiliary hopper.	Combustion poor; cinder formation. Plant running with 54 tubes plugged.
March 9	1:15	6:25	Three different coal entries tried.	Fish-tail entry gave fairly good operation for four hours and remained clean. Tube-plate leakage due to tube corrosion.
March 10	0:47	0:56	Two different designs of coal entry tried.	Difficulties with feed-screw: coal rate erratic.
March 11	0:12	4:35	Multiple air-jet entry tried.	Further difficulties with feed-screw: coal sticking in hopper. Increasing tube-plate water leakage; 17 more tubes plugged after test.
March 30	0:14	3:36	Conical coal-entry tried. Pneumatic 'bags' installed in hopper to improve feed.	Coal rate still unsteady. Large carryover; cinders blocking slag hole. Plant running with 71 tubes plugged.
March 31	0:07	2:01	Babcock and Wilcox type entry tried.	Improved combustion but coal feed still erratic.
April 1	0:12	0:54	Conical coal-entry used again.	Combustion good, but heavy HX leakage and large number of tubes plugged require 1,830°F HX inlet to give 10,000 rpm; 73 additional tubes found leaking after the test, making a total of 144 defective tubes.
			End of First Series of Tests	
	83:15	230:50	Total running time (exclusive of motoring and assisted operation).	

			Second Series of Tests	
1956 January 4-9	—	—	23:43 hours assisted running for calibration tests and drying out of insulation.	
January 9	0:59	5:54	T_{50} temperatures kept at about 1,560°F (850°C) during most of the test.	Combustion not very good: large slag formation at cyclone throat. Poor flushing of slag.
January 12-13	0:41	4:55	Various designs of coal entries tried (including thermosiphon cooling on one of them).	As above, with the addition of frequent jamming of coal-feed screw.
January 18-20	1:03	28:09	Extended water-cooled cyclone throat installed; various designs of slag drain tried.	Slag-flushing poor; drain deteriorating; very little slag flow through drain, but large accumulation of slag in mixing section. HX resistance factor up. Coal-entry cooling failure. Four HX tubes plugged after test.
February 5-6	0:32	17:01	Various coal entries tried. Slag screen installed in mixing section. Automatic icicle breaker in slag box plus 'classifier' to separate small slag pellets from icicles.	Combustion poor; cinder formation; icicle formation in slag box caused blocking. After test, Rotoclone bearing found to have failed.
February 23-24	2:41	9:50	Slag-box 'classifier' removed and all slag flushed through one opening. New coal entry design tried with additional fluidizing air on feed-screw. Stainless-steel-plate slag drain.	Feed-screw jamming, or running empty, or cycling; 12,000 rpm achieved without exceeding 1,470°F (850°C) at HX inlet.
February 29-March 3	1:27	31:09	Coal-drying system installed (uses hot air bled from turbine exhaust). Continued experimentation with slag-flushing, draining, coal entries.	Difficulties with slag-draining and flushing. Slag formation in mixing section. Coal-drying eliminated most of feed-screw difficulties. Four HX tubes plugged after test.
March 7-8	0:13	30:46	Air jets in hopper installed to insure steady feed. New inserts installed in HX.	Good run: no tube-plate blockage, but slag screen in mixing section became blocked. Slag hole open but draining poor: large slag formation from cyclone throat. Three more tubes plugged.
March 14-17	1:26	40:19	New coal entry design: 7-inch swirl-cup plus target; four cyclone-throat air jets installed. Coal entry changed after 24 hours—conical design without target.	Cooling-water flow to target failed after 10 hours. Target gradually burned away and result was combustion near cyclone throat and poor slag flow with slag build-up at throat. Combustion poor with new coal entry; tube-plate deposits heavy. Three more tubes plugged after test and four leaking tubes replaced. Total of inoperative tubes now 10.

Date of Test	Running Time (hours and minutes)		Description of Test or Modifications	Remarks
	On Oil	On Coal		
1956—con.				
April 22-25	0:36	49:57	Slag hole relocated closer to throat and to left of centre line. Swirl-cup-type coal entry.	Great improvement in slag-draining and combustion, but erratic coal feed made flame front move downstream periodically and cause cinder formation. Swirl-cup erosion caused gradual deterioration of performance.
April 25-26	0:04	51:18	Continuation of run of April 22 after 3-hour stop to install new coal entry (5-inch diameter swirl-cup). Stopped after completing 100 hours.	Slag hole blocked at times but could be opened with poker without difficulty. Coal feed still unsteady (because coal stuck in hopper and feed-screw ran empty). HX resistance factor up 60 per cent at end of test. Four tubes plugged.
May 21-24	2:00	61:28	New coal entry tried. Modifications to air ports in mixing section.	Operation generally good except for continuous trouble with coal feed. One coal entry burnt out and cyclone flooded. Restarted without difficulty after water drained. Eight tubes plugged.
June 13-14	0:26	16:26	Five new kinds of coal entries with interchangeable swirl chambers of different diameters. New sloped slag screen in mixing chamber; new screen in front of tube plate. Vibrator installed on hopper wall.	Coal feed still erratic; otherwise, operation good.
June 21-26	0:04	121:29	Scale hopper modified (aluminum sheet lining, steeper walls in lower section). Flexible connection in coal-carrying pipe. Diameter of injector forcing-air pipe increased from $\frac{5}{16}$ to $\frac{3}{8}$ inch.	100-hour non-stop run achieved for first time. Increased periods of steady operation, although feed-screw troubles not eliminated completely. Slag-drain plate dislodged by poking; mixing-section screen burnt through; 17 additional tubes found leaking.
			Heat exchanger removed for modifications; 65 tubes replaced with 57 Type 446 and 8 Nimonic (welded to 7 feet of carbon-steel tube); 74 stainless steel inserts installed.	Total time to date in second series of tests: on oil—12:12; on coal—468:41; total—480:53.

October 9-15	0:10	134:05	5-inch-diameter water-cooled coal entry with spring-steel wear strip in swirl chamber.	Long periods of steady operation, although occasional unsteadiness in coal feed still present. Stopped after 91 hours to clean mixing zone screen. Final stop necessitated by Dart-compressor failure. Eight HX tubes plugged after test.
October 31-November 11	0:25	247:24	23-inch-diameter S.S. target installed $4\frac{1}{2}$ feet downstream from cyclone throat and on its centerline. New mixing-section screen installed in two staggered sections. Dart 16 replaced by Dart 13.	Test terminated after completion of more than 200 hours non-stop. Quick stop-start test made after five hours. Rotation of turbine ceased two minutes after 'fuel off'. Self-sustained two minutes after relighting, following 17-minute stop. HX inlet temperature at start 1534°F (835°C) at 11,500 rpm; 39 tubes found leaking after test (total 47).
November 22	0:03	4:29	Demonstration of rapid starting and stopping for delegates to Conference on Coal-burning Gas Turbines.	Corrosion of remaining Nimonic tubes very rapid; 35 more tubes found leaking after this short test.
December 4	0:32	1:04	82 tubes plugged for this test.	Excessive T_{50} temperatures due to tube leakage; 22 more Nimonic tubes leaking after test. Most other alloy tubes showing no corrosion damage.
			End of Second Series of Tests	
	13:22	855:43	Total running time during second series.	

Appendix 8

OVER-ALL HEAT-TRANSFER COEFFICIENT AS A FUNCTION OF THE INDIVIDUAL REYNOLDS NUMBERS

The over-all heat-transfer coefficient U can be expressed as a function of the individual thermal resistances referred to the outside tube surface.

$$\frac{1}{U} = \frac{1}{h_o} + r_{fo} + r_w + r_{fi}\frac{d_o}{d_i} + \frac{1}{h_i}\frac{d_o}{d_i} \quad \text{[1]}$$

where U = over-all heat-transfer coefficient referred to outside tube surface
h_o = film coefficient outside tubes
h_i = film coefficient inside tubes
r_{fo} = scale resistance outside tubes
r_{fi} = scale resistance inside tubes
r_w = tube-wall resistance
d_o, d_i = tube diameters

Since both h_o and h_i are relatively low, the scale and tube-wall resistances are much smaller than the film resistances and can be neglected

Thus. $$\frac{1}{U} = \frac{1}{h_o} + \frac{1}{h_i}\frac{d_o}{d_i} \quad \text{[2]}$$

Assuming that the flow on the shell side is essentially along the tubes, the following general equation of heat transfer can be used both inside and outside the tubes.

$$Nu = 0.023\ Re^{0.8}\ Pr^{0.333} \quad \text{[3]}$$

Hence

$$h = \frac{.023\ k}{D} Re^{0.8}\ Pr^{0.333}$$

Since the Prandtl number is constant over a wide range of temperatures, and the variation of thermal conductivity k is small in the range of average temperatures usually encountered (being of the order of 10 per cent), we can simplify the foregoing equation to

$$h_{air} = k_1\ Re_a^{0.8}$$

$$h_{gas} = k_2\ Re_g^{0.8}$$

Hence $$\frac{1}{U} = \frac{1}{k_1\ Re_a^{0.8}} + \frac{1}{k_2\ Re_g^{0.8}} \quad \text{[4]}$$

Constants k_1 and k_2 are not equal, but even if this difference is neglected and $\frac{1}{U}$ is plotted against $\frac{1}{Re_a^{0.8}} + \frac{1}{Re_g^{0.8}}$, a fair indication of the dependence of the over-all coefficient on the individual Reynolds numbers can be obtained.

Appendix 9

SUMMARY OF PERFORMANCE OF THE HEAT-EXCHANGER TUBES

A summary of performance of various heat-exchanger tubes is given in this Appendix. The nominal (or actual, if available) composition of the special alloy tubes is given in the table on page 157, and their location in the bundle is shown in Figure A9-1.

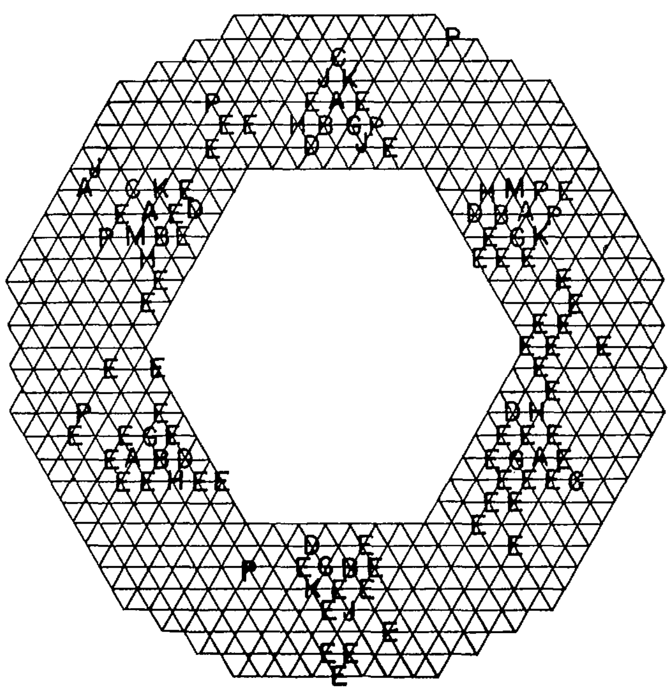

Figure A9-1. Position of special alloy tubes in air heater, September 1956.

Symbol	Type	Number Installed
A	Nimonic 75 seamless	6
B	F.C.B. (347)	5
C	Type 442	3
D	Type 310	6
E	Type 446	59
G	Incoloy	5
H	ImmV(309)	5
J	Type 321	4
K	Type 347	4
M	Type 316	2
P	Nimonic 75 welded	8
Nimonic 75 welded—remainder		

Figure A9-2 shows the location of the tubes which were examined and whose photographs appear in the report.

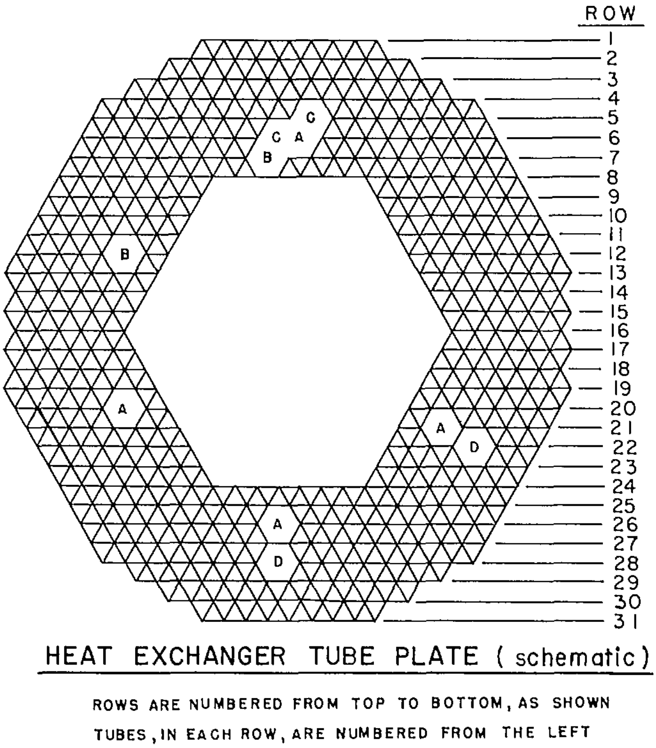

Figure A9-2. Location of alloy tubes removed for examination. Heat-exchanger tube plate. Each intersection represents the location of a tube.

A Incoloy
- row 6 tube 11 (B.W. 16-gauge)
- row 20 tube 6 (B.W. 16-gauge)
- row 21 Tube 11 (B.W. 16-gauge)
- row 26 tube 10 (B.W. 16-gauge)

B Type 309, row 12 tube 6 (B.W. 16-gauge)
Type 310, row 7 tube 10 (B.W. 20-gauge)

C Type 347, row 5 tube 11 (B.W. 20-gauge)
Type 347, row 6 tube 10 (B.W. 16-gauge)

D Type 447, row 22 tube 13 (B.W. 16-gauge)
Type 446, row 28 tube 9 (B.W. 16 gauge)

Summary of Nimonic 75 Tube Failures

First series of tests

Number of Nimonic 75 tubes installed: 498

Total hours	196	217	229	314
Total failures	48	54	71	144

Second series of tests

Number of Nimonic 75 tubes: 442 at beginning; 306 at 481 Hrs.[54]

Total hours....................	177	279	342	481	615	863	867	869
Total failures..................	1	2	7	35	43	82	116	136

Twelve failures were due to miscellaneous causes; all others resulted from corrosion.

Details of Tube Failures

(Second series of tests)

List of failures and replacements

Total Hours (oil + coal)	Number of Failures	Alloy Type	Wall Thickness	Remarks
42	2	410	.035	collapsed
	2	442	.035	collapsed
59	1	442		collapsed
	1	410		collapsed
104	1	442		collapsed
	1	304	.035	pulled out
135	2	442		collapsed
	1	410		collapsed
177	1	Ni-75	.035	leaking
	1	442		collapsed
	1	410		collapsed

Removed 5 tubes—submitted to Dept. Mines & Technical Surveys for examination;
—replaced by Nimonic 75 tubes.

Total Hours (oil + coal)	Number of Failures	Alloy Type	Remarks
279	1	Nimonic 75	leaking
	2	442	collapsed
	1	442	collapsed (not plugged)
342	5	Nimonic 75	leaking
	1	304	leaking
	1	316	leaking
	1	321	leaking
481	13	Nimonic 75	leaking
	3	Nimonic 75	obstructed
	9	347	pulled out

Removed 65 tubes—submitted to Dept. Mines & Technical Surveys for examination;
—replaced by 57 type 446 tubes and 8 Nimonic 75 tubes welded to 7-foot lengths of carbon-steel tubes.

Total Hours (oil + coal)	Number of Failures	Type of Alloy	Remarks
615	8	Nimonic 75	leaking
863	39	Nimonic 75	leaking
867	34	Nimonic 75	leaking
	1	321	leaking
869	20	Ninomic 75	
	1	321	leaking
	1	347	leaking

[54] All defective tubes, including some with expansion trouble, were removed at 481 hours, and 306 Nimonic 75 were in use at the beginning of the next phase of testing.

History of Special Alloy Tubes

Incoloy	5 tubes installed—no failures
AISI 310	6 tubes installed—no failures
IMMV (309)	5 tubes installed—no failures
AISI 347	6 tubes installed 2 tubes removed at 481 hours—expansion trouble 1 tube removed at 869 hours—leaked
FCB (T)	5 tubes—no failures
AISI 321	11 tubes installed 1 tube failed by leaking at 342 hours 6 tubes had expansion trouble at 481 hours 1 tube failed by leaking at 867 hours 1 tube failed by leaking at 869 hours
AISI 316	1 tube failed by leaking at 342 hours 1 tube had expansion trouble at 481 hours
AISI 304	6 tubes installed 1 tube had expansion trouble at 104 hours 1 tube leaked at 342 hours 4 tubes had expansion trouble at 481 hours
AISI 410	5 tubes installed 2 tubes collapsed at 42 hours 1 tube collapsed at 59 hours 1 tube collapsed at 135 hours 1 tube collapsed at 177 hours
AISI 442	14 tubes installed 2 tubes collapsed at 42 hours 1 tube collapsed at 59 hours 1 tube collapsed at 104 hours 2 tubes collapsed at 135 hours 1 tube collapsed at 177 hours 1 tube collapsed at 279 hours 1 tube collapsed at 481 hours 3 tubes still in use after 869 hours
AISI 446	3 tubes originally installed No failures 1 tube removed as sample at 481 hours 57 new tubes installed at same time No failures after 869 hours on original 2 tubes No failures after 388 hours on 57 new tubes

Alloys Used as Trial Tubes in Heat Exchanger

Alloy Number	Type of Alloy	Composition											
		Fe	Cr	Ni	Si	Mn	Cu	C	Ti	Mo	Nb	Ta	Balance
Nimonic 75	–	5	18–21	–	1	1.0	0.5	0.08 –0.15	0.2–0.6	–	–	–	Ni
Incoloy	–	–	19–22	32–36	–	<1.5	–	<0.10	–	–	–	–	Fe
Type 304	Austenitic stainless........	–	18–20	8–11	–	<2.0	–	<0.08	–	–	–	–	Fe
Type 309	Austenitic stainless........	–	22–24	12–15	–	<2.0	–	<0.20	–	–	–	–	Fe
Type 310	Austenitic stainless........	–	24–26	19–22	–	<2.0	–	<0.25	–	–	–	–	Fe
Type 316	Austenitic stainless........	–	16–18	10–14	–	–	–	<0.10	–	2–3	–	–	Fe
Type 321	Austenitic stainless........	–	17–19	8–11	–	–	–	<0.08	More than 5xC	–	–	–	Fe
Type 347	Austenitic stainless........	–	(18) 17–19	(8) 9–12	–	–	–	<0.08	–	–	More than 10xC	Present	Fe
Type 410	Martensitic stainless.......	–	(13) 11.5–13.5	–	–	–	–	<0.15	–	–	–	–	Fe
Type 442	Ferritic stainless...........	–	(20) 18–23	–	–	–	–	<0.20	–	–	–	–	Fe
Type 446	Ferritic stainless...........	–	(27) 23–27	–	–	–	–	<0.35	–	–	–	–	Fe

(N.B. The actual figures, where available, are given in parentheses.)

Appendix 10

THERMODYNAMIC ANALYSIS OF RESULTS OBTAINED ON TEST SCHEDULE No. Q 55-2, May 21 to 24, 1956

Composition of Furnace Exhaust Gases

Air supplied to furnace[55] $13.97 \times .972 = 13.57$ lb/sec
Fuel supplied to furnace $= 0.267$ lb/sec

Fuel composition

C	—74.5% (75% less 5% combustible in slag and ash)
H_2	— 5.2%
S	— 3.2%
N_2	— 1.4%
O_2	— 4.8%
Ash	— 9.9% (9.4% plus 5% combustible in slag)
Moisture	— 1.0%

Oxygen required for combustion

.745 lb C requires $.745 \frac{32}{12} = 1.988$ lb O_2 to give $.745 \frac{45}{12} = 2.728$ lb CO_2

.052 lb H_2 requires $.052 \times 8 = .416$ lb O_2 to give $.052 \times 9 = .468$ lb H_2O

.032 lb S requires $.032 \times 1 = .032$ lb O_2 to give $.032 \times 2 = .064$ lb SO_2

O_2 required per lb of fuel	2.436 lb
O_2 present in coal	.048 lb
O_2 required from air	2.388 lb

Air required per second $\frac{2.388}{.231} \times .276 = 2.852$ lb

Excess air present in gas $13.57 - 2.85 = 10.72$ lb

Exhaust-gas composition

				lb/sec	
CO_2		$2.728 \times .276$		0.753	5.45%
H_2O	from combustion	$.468 \times .276$		0.129	0.93%
	from moisture	$.010 \times .276$		0.003	0.02%
SO_2		$.064 \times .276$		0.018	0.13%
N_2	from combustion air	$2.852 \times .769$	$= 2.193$		
	from excess air	$10.72 \times .769$	$= 8.245$		
	present in fuel	$.014 \times .267$	$= .003$	10.441	75.55%
O_2	from excess air	$10.72 \times .231$		2.477	17.92%
Total exhaust-gas flow				13.821 lb/sec	

[55] Figure A 10-1 shows the flows through various parts of the plant.

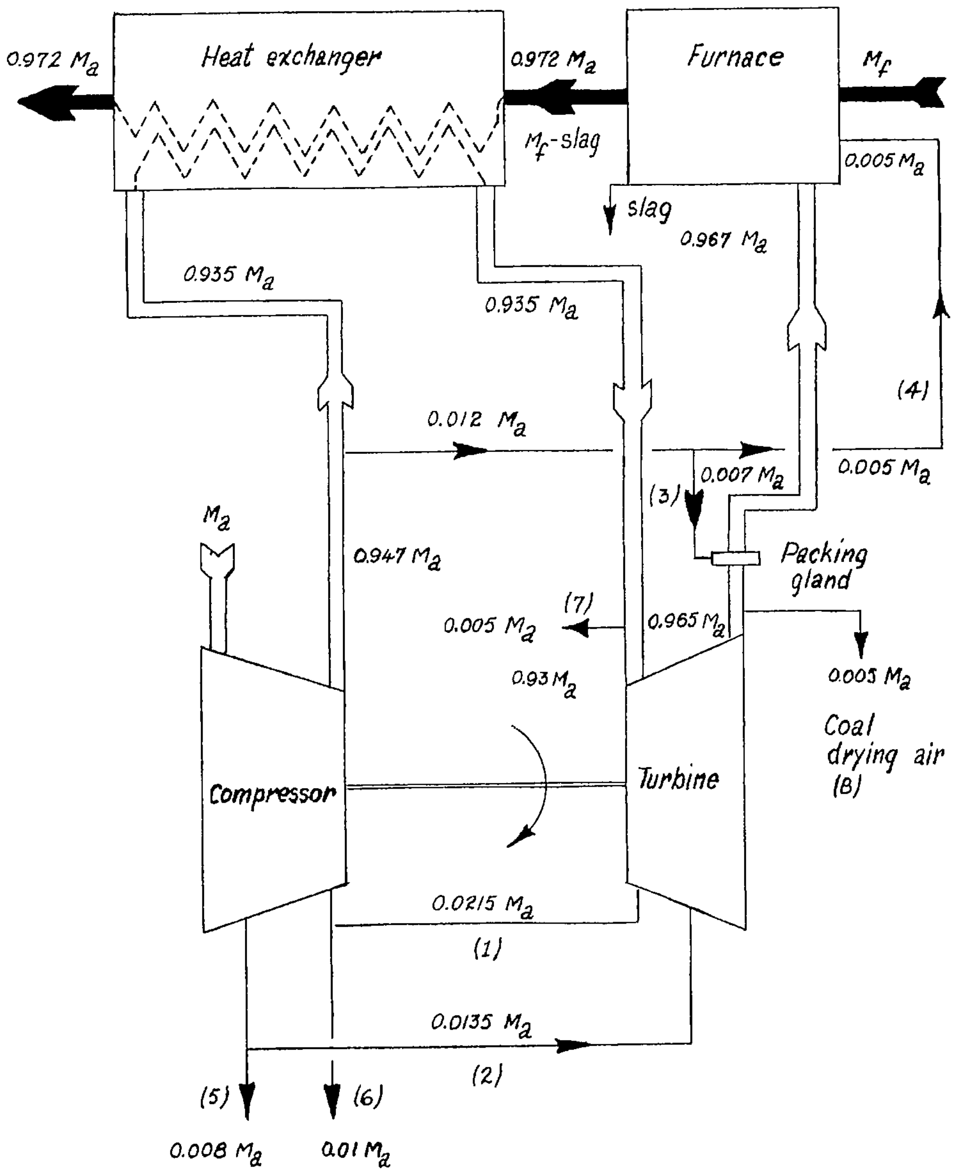

Figure A10-1. Air- and fuel-flow diagram.

Furnace

(coal rate .276 lb/sec)

Constituent	lb/sec	Inlet °C	Outlet °C	ΔT	C_p	Chu/sec	% of Total Heat Release
CO_2	.753	408	828	420	.288	91	
O_2 and SO_2	2.495	408	828	420	.255	268	
N_2	10.441	408	828	420	.272	1,194	
H_2O evaporation + superheat	.132	50	828	778		119	
Tapped slag	.028	50	1,200	1,150	.165	5	
Heat applied usefully						1,677	81.2
Cyclone cooling water	.41	10	89.5	79.5	1.0	326	
Service cooling water						20	
Convection and radiation heat loss						41	
Heat losses						387	18.8
Total heat released by coal						2,064	100.0

$$\text{Heat released per pound of coal gross} = 2064 \div .276$$
$$= 7{,}480 \text{ Chu}$$
$$= 13{,}480 \text{ Btu}$$

Heater

Heat Balance	lb/sec	°C Inlet	°C Out	ΔT	C_p	Chu/sec	% of Total Heat Release
Gas pass							
Gas flow	13.82	828	391	437	.272	1,642	
Air pass							
Air flow (93.5%)	13.06	154	612	458	.254	1,517	73.2
Heat loss from shell						31	1.6
Total heat gained by air						1,548	74.8
Tube-plate cooling water	4.1	55	78	23	1.0	94	4.5
Total heat lost by gas						1,642	79.3

Exhaust-Duct Loss

Gas flow 13.82 lb/sec

Convection and radiation loss obtained from heat balance on test-cell ventilating air 23 Chu/sec

Inlet temperature 391°C

Mean specific heat .255

∴ outlet temperature 385°C

Exhaust-Stack Loss

ASME method	Stack inlet temperature	385°C
	Plant inlet temperature	10°C
	Temperature rise	375°C

	lb/sec	C_p	T	Loss Chu/sec	
Dry gas	13.69	.247	375	1,264	
Moisture	$0.132 \times \left(592 + \frac{375}{2} - 10\right)$			102	
Total stack loss				1,366	66%

Compressor

Inlet conditions	14.7 lb/in², 10°C
Total mass-flow	13.97 lb/sec
Compression ratio	3.265
Power-input adiabatic efficiency	74.0
Measured-temperature-rise adiabatic efficiency	78.9

See Rolls-Royce compressor test S 4240/1118.

$$\frac{M\sqrt{T}}{P} = 15.88$$

$$\frac{HP}{P\sqrt{T}} = 5.27$$

$$\frac{U}{\sqrt{T}} = 79.4$$

Heat Balance

	lb/sec	Inlet °C	Outlet °C	Rise °C	Specific Heat	Chu/sec
Main air flow 98.65%	13.78	10	154	144	.242	480
LP cooling air 1.35%	.19	10	85	75	.24	3
Heat loss						30
Power to drive compressor						513

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Turbine

So that the turbine efficiency may be estimated, the heat losses in the turbine have been divided into those occurring before and those occurring after the conversion to mechanical work. It has thus been possible to estimate the temperature from which the mechanical work was done and the turbine efficiency.

Losses from turbine and related ducting

	lb/sec	Inlet °C	Out °C	ΔT	C_p	Chu/sec	
Convection heat loss........						30	before work 67
Expansion-joint cooling water						27	
HP cooling air.............	.30	154	417	263	.249	10	
						10	after work 46
LP cooling air.............	.189	75	417	342	.247	16	
Convection heat loss........						20	

	Chu/sec	Air Inlet °C	Air C_p	Air lb/sec	Air ΔT	Air Outlet °C
Losses before work...............	67	612	.266	13.0	18	594
Work stage						
To compressor.................	513	594	.2615	13.0	150	
To useful power...............	44				13	
Heat to oil..................	6				2	
	563				165	429
Losses after work...............	46	429	.257	13.0	14	415

Turbine-Furnace Duct

Convection heat loss (15% of total ventilation loss) = 30 Chu/sec

	Chu/sec	Air-inlet Temperature °C	lb/sec	T °C	Outlet °C
Heat loss.........................	30	415	13.52	8.7	407

Ventilation Loss

(obtained by heat balance on flow of ventilating air through test cell)

Mass flow..................	34.1	lb/sec
Temperature rise............	25	°C
Mean C_p..................	.24	Chu/lb°C
Heat loss..................	205	Chu/sec or 9.9% of total heat release

Distribution of losses was based on surface areas and temperatures of various components of the plant and the following breakdown of losses was arrived at:

Component	Loss Chu/sec	% of Total Heat Release
Furnace	41	1.98
Heater	31	1.50
Turbine inlet duct	30	1.45
Furnace inlet duct	30	1.45
Compressor and outlet duct	30	1.45
Exhaust duct	23	1.11
Turbine	20	.96
	205	9.9

Cooling-Water Losses

Cooling cyclone	326	15.7
Miscellaneous furnace	22	1.1
Oil cooler	6	.3
Heater tube plate	94	4.5
Turbine slip joints, etc	20	1.0
	468	22.6

Miscellaneous losses: Coal-drying, 3 Chu/sec.

Heat Balance on Complete Plant

Item	Chu/sec	Btu/sec	% of Heat Supplied in Coal
Heat to useful power	44	79	2.13
Heat to cooling water[56]	468	842	22.6
Heat to exhaust	1,366	2,458	66.0
Conviction and radiation loss	205	369	9.9
Miscellaneous (coal drying)	3	6	.15
Total	2,086	3,754	100.8
(Unaccounted for)	(−22)		(−0.8)

The "percentage of heat supplied in coal" column in the foregoing table is based on the average calorific value obtained on five samples of coal used in this test. A value of 13,500 Btu/lb was used as the higher heating value of the coal.

[56] This includes the heat picked up in the oil cooler.

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Appendix 11

ESTIMATE OF PERFORMANCE OF LARGER AIR HEATER

Nomenclature

A = Surface area, sq ft
C_p = Specific heat at constant pressure Chu/lb °C or Btu/lb °F
D_e = Equivalent diameter
= 4 × flow area/wetted perimeter
D_i = Tube inside diameter, ft
D_o = Tube outside diameter, ft
f = Friction factor in Fanning Equation
G = Mass velocity, lb/hr ft²
k = Thermal conductivity Chu/hr ft °C
$K_1, K_2, K_3, K_4, K_5, K_6, K_7, K_8, K_9$ = Constants
L = Tube length, ft
N = Number of tubes in shell
ΔP = Pressure drop, psi
Q = Heat transferred, Chu/hr
t^*_{50} = Gas temperature at heat-exchanger inlet, allowing for heat lost to tube-plate coolant, °C
t_{70} = Gas temperature at heat-exchanger outlet, °C
t_{30} = Air temperature at heat-exchanger inlet, °C
t_{20} = Air temperature at heat-exchanger outlet, (turbine inlet), °C
ΔT = Log-mean-temperature difference, °C
U = Over-all heat-transfer coefficient Chu/hr ft² °C
W = Weight flow, lb/hr
ρ = Density, lb/cu ft
μ = Viscosity, lb/ft hr

Subscripts

a = Air
g = Gas
1 = Actual heater used
2 = Heater with increased area

The original exhaust-heated-cycle arrangement tested in the Gas Dynamics Laboratory featured two heat exchangers. For the second series of tests, however, the cold exchanger was removed, the available heat transfer surface being thus reduced from a total of 4,900 to 2,390 square feet.

The reduction of the heat-transfer surface caused an increase of the final exhaust temperature and a lowering of the turbine-inlet temperature, thus reducing the efficiency of the plant.

Had it been possible to build a new and larger heater, a much better performance could have been obtained. The calculations given in this Appendix show the increase in performance of the exchanger which could have been obtained with a heater having twice the surface area of the hot exchanger (or an area approximately equal to the total surface available during the first series of tests).

Assumptions

(1) The new heater is similar in design to the one actually used and has twice as much heat-transfer surface (4,780 square feet).

(2) The length and diameter of the new heater is such that the pressure drop through it is exactly the same as that obtained during the tests with the existing exchanger.

(3) The flows and air and gas inlet-temperatures are unchanged from their test values.

(4) Heat-transfer coefficients are given by the following expression on both the shell and the tube sides.

$$h = \frac{.023\ k}{D_e}\left(\frac{G\ D_e}{\mu}\right)^{0.8} Pr^{1/3} \qquad [1]$$

(5) Pressure drops are given by the following expression on both the shell and the tube side:

$$\Delta P = \frac{f\ G^2\ L}{K_1\,\rho\,D_e} \qquad [2a]$$

$$\text{where } f = \frac{K_2}{\left(\dfrac{GD_e}{\mu}\right)^{0.2}} \qquad [2b]$$

Consider the pressure-drop equations.

Substituting [2b] into [2a] we obtain:

$$\Delta P = K_3\ G^{1.8}\ L \qquad [3]$$

$$\text{where } K_3 = \frac{K_2 \times \mu^{0.2}}{K_1\ \mu\ (D_e)^{1.2}}$$

$$\text{But } G = \frac{W}{\text{flow area}} = \frac{K_4}{N} \qquad [4]$$

since the flow area is proportional to the number of tubes (this is true on both the shell and the tube sides).

Thus equation [3] becomes

$$\Delta P = K_5\,\frac{L}{N^{1.8}} \qquad [5]$$

where $K_5 = K_3\ K_4^{1.8}$

Let the length and number of tubes in the actual heater be L_1 and N_1 respectively, and in the larger heater L_2 and N_2.

Then, since the pressure drop is to remain unchanged,

$$\Delta P_1 = \Delta P_2 \therefore \frac{L_1}{N_1^{1.8}} = \frac{L_2}{N_2^{1.8}}$$

$$\text{or } \frac{L_1}{L_2} = \left(\frac{N_1}{N_2}\right)^{1.8} \qquad [6]$$

Now the surface area of each exchanger is given by

$$A_s = \pi D_o N \times L = K_6 \times N L$$

Since the second exchanger is to have double the surface of the first,

$$\frac{N_2 L_2}{N_1 L_1} = 2.0 \qquad [7]$$

Solution of simultaneous equations [6] and [7] gives

$$\frac{N_2}{N_1} = 1.28$$

$$\frac{L_2}{L_1} = 1.56$$

But $\quad N_1 = 498, \; L_1 = 18.1$ ft

$\therefore \quad N_2 = 637, \; L_2 = 28.2$ ft

Now consider the heat-transfer coefficients.

By grouping all the constant terms together, equation [1] can be simplified to:

$$h_g = \frac{K_7}{N^{0.8}}, \quad h_a = \frac{K_8}{N^{0.8}}$$

But
$$\frac{1}{U} = \frac{1}{h_a} + \frac{1}{h_g}\frac{D_o}{D_i}$$

thus
$$\frac{1}{U} = N^{0.8}\left[\frac{1}{K_8} + \frac{1}{K_8}\frac{D_o}{D_i}\right]$$

$$U = \frac{K_9}{N^{0.8}} \qquad [8]$$

The results of the test of May 21, 1956, (*see* Appendix 12) give

$$t_{50}^* = 803.6\ ^\circ\mathrm{C}$$

$$t_{20} = 154\ ^\circ\mathrm{C}$$

$$U_1 = 11.2$$

Consequently,

$$U_2 = U_1 \times \left(\frac{N_1}{N_2}\right)^{0.8}$$

$$= 11.2 \times \left(\frac{498}{637}\right)^{0.8} = 9.2 \qquad [9]$$

The total heat transferred is:

$$Q_2 = U_2 A_2 \Delta T_2 = 9.2 \times 4780 \Delta T_2 = 440{,}000 \Delta T_2 \qquad [10]$$

Also $$Q_2 = W_g Cp_g (t_{50}^* - t_{70}) \qquad [11]$$

and $$Q_2 = W_a Cp_a (t_{30} - t_{20}) \qquad [12]$$

But, from Appendix 12,

$$W_g Cp_g = 13.83 \times 3{,}600 \times .266 = 13{,}250 \text{ Chu/hr°C}$$
$$W_a Cp_a = 13.06 \times 3{,}600 \times .254 = 11{,}950 \text{ Chu/hr°C}$$

Also, $$\Delta T_2 = \frac{(t_{50}^* - t_{30}) - (t_{70} - t_{20})}{\dfrac{(t_{50} - t_{20})}{(t_{70} - t_{20})}} \qquad [13]$$

Substituting values of $W C_p$ into equations [11] and [12] and solving the four simultaneous equations [10], [11], [12], and [13] gives finally

$$Q_2 = 6.345 \times 10^6 \text{ Chu/hr.}$$

$$t_{20} = 684 \text{ °C (1262 °F)}$$

$$t_{70} = 325 \text{ °C (617 °F)}$$

Thus the turbine-inlet temperature t_{20} is seen to increase to 684°C from the 612°C obtained during the tests. Similarly, the exhaust temperature drops to 325°C from 391°C, reducing the heat lost in exhaust by more than 17 per cent.

Both these effects (higher turbine temperature and lower exhaust temperature) would have a very pronounced effect on the power output and efficiency of the unit.

Appendix 12

DERIVATION OF TUBE-TEMPERATURE EQUATION

The derivation of this equation is based on the fact that most of the running during the second series of tests was done under fairly constant conditions of speed and temperature.

Since gas and air flows and temperatures varied within very narrow limits, the film coefficients could be regarded as essentially constant, the final expression being thus considerably simplified.

The heat lost by the gas to the tube-plate coolant was allowed for by reducing the measured gas-inlet temperature by a factor obtained from actual measurement of the tube-plate cooling losses.

Assumptions Made in Deriving the Equation

Assumption 1—The heat-transfer coefficients for all operating conditions are constant along the full length of the tube.

This condition is, of course, not fulfilled in practice. Departure from it will not, however, result in a large error if both shell- and tube-side coefficients behave similarly, since the tube-wall temperature depends on their *ratio*.

Since essentially the same fluid stream flows through both sides of the exchanger, any increase in flow rate or temperature level on one side of the exchanger will immediately result in a similar increase on the other side. Thus, even though the film coefficients may change appreciably, their ratio will remain practically constant.

Similar reasoning can be applied to the variation of film coefficients along the length of the tube. That this variation is not great is illustrated in the following table.

Variation of Film Coefficient with Distance from Hot End

Position	Gas (tube side)			Air (shell side)		
	Hot End	Middle	Cold End	Hot End	Middle	Cold End
Typical temperature[57] °F	1,480[58]	1,106	736	1,134	721	306
Flow lb./sec.		13.83			13.06	
Film coefficient h	25.8	24.1	22.6	28.5	25.7	22.6
Difference from h_{ave}	+6%	0	−7%	+11%	0	−12%
h_{air}/h_{gas}				1.10	1.06	1.00

The foregoing table shows that, although film coefficients may vary by as much as 23 per cent, the maximum variation of their ratio is only 10 per cent, and the variation between the hot end and the average is only 4 per cent.

[57] Taken from test of May 21-24, 1956.
[58] Allowing for tube-plate coolant loss.

Since the film coefficients used in the derivation of the tube-temperature formula were computed at average temperatures, the error at the hot end would be that due to using $h_a/h_g = 1.06$ instead of 1.10. A simple calculation will reveal that this results in an error of only 4°F on the high (safe) side.

The coefficients quoted in this table were computed by using the tube-side correlation with an appropriate equivalent diameter on the shell side.

Since there is considerable doubt about the true nature of flow on the shell side, h_{air} used in the derivation of the tube-temperature equation was obtained by calculating the tube-side coefficient h_{gas} and then making use of the following equation to find h_{air}:

$$h_a = \frac{U}{1 - \dfrac{U}{h_g} \times \dfrac{d_o}{d_i}}$$

where U is the experimentally determined over-all coefficient.

This gives an average coefficient h_a for the full length of the exchanger. It may be argued that near the front tube plate, where the air crosses the bundle in order to reach the outlet nozzles, h_a will be higher. This may be true, but previous experience[59] has shown that the reverse may also be true (particularly with a three-nozzle outlet arrangement, in which the flow area increases appreciably near the outside of the bundle).

In view of this uncertainty, it was impossible to make an allowance which could be regarded as at all accurate and a constant shell-side coefficient was used for the full tube length.

Assumption 2—The ratios $\frac{U}{h_a}$ and $\frac{W_a C_{pa}}{W_g C_{pg}}$ are constant for all conditions normally encountered during the tests.

This assumption is very nearly true in view of the arguments given in the discussion of Assumption 1 (ratios are again being dealt with), and in view of the constancy of conditions during most of the tests.

Assumption 3—To simplify the calculations it is assumed that temperatures in the exchanger vary linearly with distance along the tubes. The validity of this assumption is discussed fully on pages 173-175.

Assumption 4—The air temperature at the inlet to the exchanger (t_{20}) remains constant at 154°C.

This is very nearly true for all tests run at about 11,000 rpm and normal ambient air temperatures. On extremely cold winter days, when t_{20} is considerably lower, a more accurate result can be obtained by using the actual value of t_{20} in Equation [9].

Assumption 5—All distances along the tubes are measured from a point 2 inches downstream from the rear face of the main tube-sheet.

Experimental evidence pointed to the fact that this is the point of maximum temperature, just outside the region where tube-plate cooling effects predominate.

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[59] See Figure 30b for the coefficient in the parallel-flow section of the first exchanger. This section was arranged for nearly pure cross-flow on the shell side.

Average Test Results

Typical test results, considered representative of most of the running during the second series of tests, were taken from the test of May 21-24, 1956.

Air side

Mass flow W_a	13.06 lb/sec
Inlet temperature t_{20}	154°C
Outlet temperature t_{30}	612°C
Temperature rise t_{30}-t_{20}	458°C
Average temperature	383°C
Average specific heat Cp_a	.254 Chu/lb °C

Gas side

Mass flow	13.83 lb/sec
Inlet temperature (observed) t_{50}	828°C
Outlet temperature t_{70}	391°C
Tube-plate cooling-water flow	4.1 lb/sec
Cooling-water inlet temperature	55°C
Cooling-water outlet temperature	78°C
Heat lost to cooling water	94 Chu/sec

Temperature drop through tube plate

Cp_g (of gas) at 816°C . . . . .277 Chu/lb

$$\text{Gas-temperature drop} = \frac{94}{13.83 \times .277} = 24.4°\text{C}$$

$$\text{Corrected gas-inlet temperature } t^*_{50} = 803.6°\text{C}$$

Thermal effectiveness (corrected)

$$\eta_{th} = \frac{t_{30} - t_{20}}{t^*_{50} - t_{20}} = \frac{458}{650} = 0.705$$

Calculation of Heat Transfer Coefficients

1. Over-all U

$$\Delta t_h = 803.6 - 612 = 191.6°\text{C}$$

$$\Delta t_c = 391 - 154 = 237°\text{C}$$

$$\text{LMTD} = \frac{\Delta t_h - \Delta t_c}{l \dfrac{\Delta t_h}{\Delta t_c}} = 213°\text{C}$$

$$Q = [803.6 - 391] \times .266 \times 13.83 = 1520 \text{ Chu/sec} = 5.47 \times 10^6 \text{ Chu/hr}$$

$$A_s = 486 \times \pi \times .0833 \times 18 = 2290 \text{ sq ft}$$[60]

$$U = \frac{Q}{A_s \times \text{LMTD}} = 11.2$$

[60] Fourteen tubes were plugged for this test.

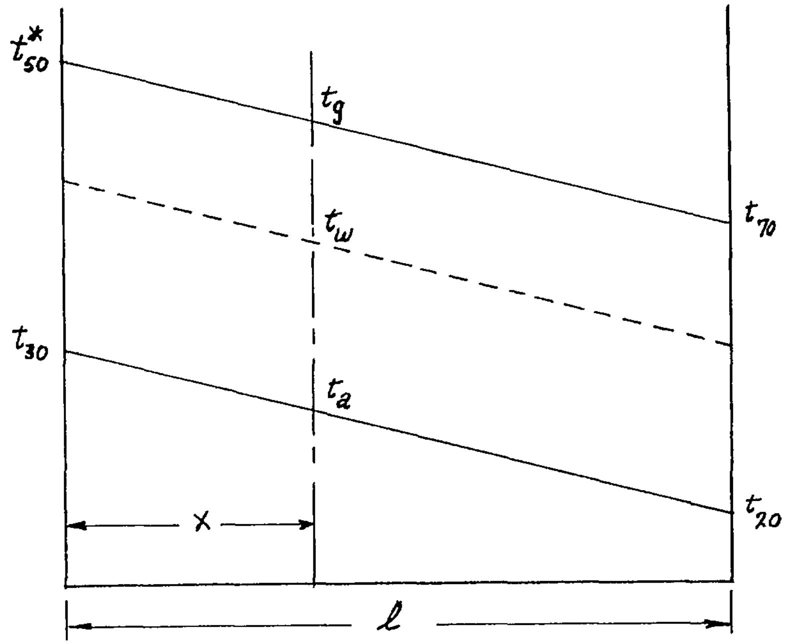

Diagram A

2. Tube side

Tube I.D. = 0.93″ = .0775′

$$\text{Flow area} = A = 486 \times \frac{.679}{144} = 2.28 \text{ sq ft}$$

$$G = \frac{W}{A} = \frac{13.83}{2.28} \times 3{,}600 = 21{,}800 \text{ lb/hr ft}^2$$

Average temperature	597°C
Viscosity	0.0938 lb/ft hr
Specific heat	.266 Chu/lb°C
Prandtl Number	0.69

$$Re_g = \frac{GD_i}{\mu} = \frac{21{,}800 \times .0775}{.0938} = 18{,}000$$

$$h_g = \frac{C_pG}{Pr_g^{2/3}} \times \frac{.023}{Re_g^{0.2}} = \frac{0.266 \times 21{,}800}{0.69^{2/3}} \times \frac{.023}{18{,}000^{0.2}} =$$

$$= 24.1 \text{ Btu/hr ft}^2{}^\circ\text{F (Chu/hr ft}^2{}^\circ\text{C)}$$

(The viscosity correction factor is close to 1.0 and is neglected here.)

3. Shell side

$$h_a = \frac{U}{1 - \frac{U}{h_g} \times \frac{d_o}{d_i}} = \frac{11.2}{1 - \frac{11.2}{24.1} \times \frac{1}{.93}} = 22.4$$

Derivation of Equation

$$t_w = t_a + \frac{h_g}{h_g + h_a}[t_g - t_a]$$

$$= t_a + \frac{U}{h_a}[t_g - t_a] \qquad [1]$$

$$\text{Also, } t_a = t_{30} - \frac{x}{l}(t_{30} - t_{20}) \qquad [2]$$

$$\text{and } t_g = t^*_{50} - \frac{x}{l}(t^*_{50} - t_{70}) \qquad [3]$$

$$\therefore t_g - t_a = (t^*_{50} - t_{30}) - \frac{x}{l}[(t^*_{50} - t_{70}) - (t_{30} - t_{20})] \qquad [4]$$

$$\therefore t_w = t_{30} - \frac{x}{l}(t_{30} - t_{20}) + \frac{U}{h_a}\left\{(t^*_{50} - t_{30}) - \frac{x}{l}[(t^*_{50} - t_{70}) - (t_{30} - t_{20})]\right\} \qquad [5]$$

Introducing the thermal effectiveness,

$$\eta_{th} = \frac{t_{30} - t_{20}}{t^*_{50} - t_{20}}$$

$$\text{Hence, } t_{30} = t_{20} + \eta_{th}(t^*_{50} - t_{20}) \qquad [6]$$

$$\text{Also, } Q = W_a Cp_a(t_{30} - t_{20}) = W_g Cp_g(t^*_{50} - t_{70}) \qquad [7]$$

Substitution of equations [6] and [7] into equation [5] gives, after some manipulation,

$$t_w = t_{20} + \left\{\eta_{th} + \frac{U}{h_a}(1 - \eta_{th}) + \frac{x\ \eta_{th}}{l}\left[\frac{U}{h_a}\left(1 - \frac{W_aCp_a}{W_gCp_g}\right) - 1\right]\right\}(t^*_{50} - t_{20}) \qquad [8]$$

Equation [8] constants are now evaluated:

$$\frac{U}{h_a} = \frac{11.2}{22.4} = 0.50 \qquad \eta_{th} = 0.705$$

$$\frac{W_aCp_a}{W_gCp_g} = \frac{13.06 \times .254}{13.83 \times .266} = 0.90$$

l = 18′ − 9.25″ total length of tubes
less 7.25″ length at tube-sheets
less 2.0″ distance from rear of hot tube-sheet to maximum temperature point
= 18′0″ effective length

Substitution in Equation [8] gives

$$t_w = t_{20} + (0.853 - 0.0372x)(t^*_{50} - t_{20}) \qquad [9]$$

Since t_{20} is usually very close to 154°C, this value can be substituted in Equation [9] to obtain:

$$t_w = t^*_{50}(.853 - .0372x) + 23 + 5.72x \qquad [10]$$

Most tests indicate that the heat lost to the tube-plate cooling water results in a reduction of the measured gas-inlet temperature (t_{50}) to a value which can be expressed as

$$t^*_{50} = t_{50} - .0294\, t_{50}$$

Substitution of this in Equation [10] gives finally

$$\underline{t_w = t_{50}\,(.828 - .036x) + 23 + 5.72x} \quad \ldots\ldots [11]$$

in which all temperatures are in °C and x is the distance in feet from the hottest point of the tubes.

Substantiation of Assumption 3

The assumption of linear temperature variation greatly simplifies the derivation of the tube-wall temperature equation without sacrificing unduly the accuracy of the final expression.

A more rigorous derivation is given here to show that the actual temperature variation along the exchanger does not depart greatly from a straight-line relation.

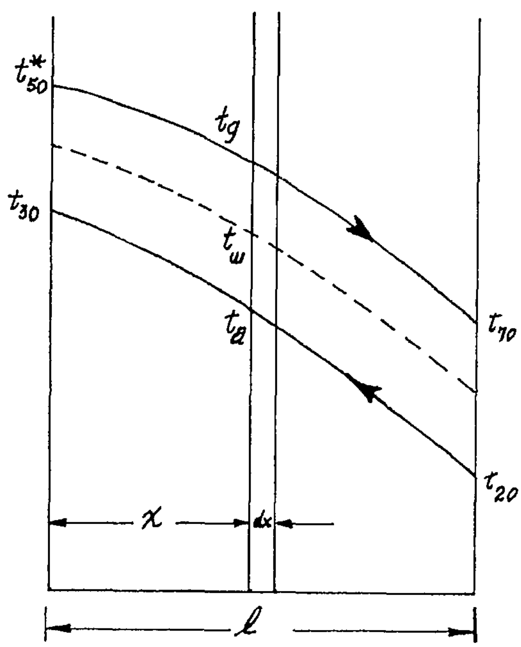

Diagram B

With reference to the diagram, the differential equation for the heat transferred across an elemental length dx of the exchanger is:

$$dQ = U\, dA\, [t_g - t_a] \quad \ldots\ldots [12]$$

or $dQ = U\ P\ [t_g - t_a]\ dx$ .... [13]

where P is the total heat-transfer surface per unit length of the tubes.

Since the heat picked up by the air is equal to the heat lost by the gas,

$$dQ = -W_a\ Cp_a\ dt_a = -W_g\ Cp_g\ dt_g \qquad [14]$$

Combined, Equations [14] and [13] give

$$dt_a = -\frac{U\ P}{W_a\ Cp_a}(t_g - t_a)\ dx \qquad [15]$$

Equation [14] when applied to the length x of the exchanger, becomes

$$W_a\ Cp_a\ [t_{30} - t_a] = W_g\ Cp_g\ [t^*_{50} - t_g]$$

Hence,

$$t_g = t_{50} - \frac{W_a\ Cp_a}{W_g\ Cp_g}[t_{30} - t_a]$$

If $\frac{W_a\ Cp_a}{W_g\ Cp_g} = R$, then $t_g = t^*_{50} - R\ [t_{30} - t_a]$ .... [16]

Substituting Equation [16] into [15] gives:

$$dt_a = -\frac{U\ P}{W_a\ Cp_a}[t^*_{50} - R(t_{30} - t_a) - t_a]\ dx$$

$$\therefore \frac{dt_a}{(t^*_{50} - Rt_{30}) + (R - 1)t_a} = -\frac{U\ P}{W_a\ Cp_a}dx$$

Integration of the temperatures between t_{30} and t_a and lengths between $x = 0$ and $x = x$ results in

$$\frac{1}{R - 1}\ln\frac{(t^*_{50} - Rt_{30}) + (R - 1)\ ta}{(t^*_{50} - Rt_{30}) + (R - 1)\ t_{30}} = -\frac{U\ P}{W_a\ Cp_a}x$$

which yields, upon elimination of logarithms,

$$\frac{(t^*_{50} - Rt_{30}) + (R - 1)\ t_a}{(t^*_{50} - t_{30})} = e^{-\frac{UP}{W_aCp_a}(R - 1)\ x}$$

Hence,

$$t_a = \frac{1}{(1 - R)}\left[(t^*_{50} - Rt_{30}) - (t^*_{50} - t_{30})\ e^{\frac{UP}{W_aCp_a}(1 - R)\ x}\right] \qquad [17]$$

By a similar procedure the gas temperature is found to be

$$t_g = \frac{1}{(1 - R)}\left[(t^*_{50} - Rt_{30}) - R(t^*_{50} - t_{30})\ e^{\frac{UP}{W_aCp_a}(1 - R)\ x}\right] \qquad [18]$$

These two equations can be combined to give the wall temperature t_w if it is remembered that

$$t_w = t_a + \frac{U}{h_a}(t_g - t_a) \quad [19]$$

and

$$t_{30} = t_{20} + \eta_{th}(t^*_{50} - t_{20}) \quad [20]$$

Thus, finally, the wall temperature becomes:

$$t_w = \frac{U}{h_a(1 - R)}\Big\{[(1 - R\eta_{th}) - R(1 - \eta_{th})e^{Kx}]t^*_{50} + + R(1 - \eta_{th})(e^{Kx} - 1)t_{20}\Big\} + \left(1 - \frac{U}{h_a}\right)\frac{1}{(1 - R)}\Big\{[(1 - R\eta_{th}) - - (1 - \eta_{th})e^{Kx}]t^*_{50} + (e^{Kx} - R)(1 - \eta_{th})t_{20}\Big\} \quad [21]$$

$$\text{where } K = \frac{UP}{W_aCp_a}(1 - R)$$

By using the heat-transfer coefficients and temperatures calculated earlier in this appendix, Equation [21] can be reduced to

$$\underline{t_w = t_{20} + (3.655 - 2.8025e^{.0119x})(t^*_{50} - t_{20})} \quad [22]$$

in which all temperatures are in °C and x is in feet.

A comparison of the results obtained with Equation [9] and Equation [22] for the case of $t^*_{50} = 803.6°C$ and $t_{20} = 154°C$ is given in Figure A12-1.

The maximum deviation occurs at a distance of 9 feet and amounts only to 13°C. Near the hot tube-sheet the deviation is insignificant.

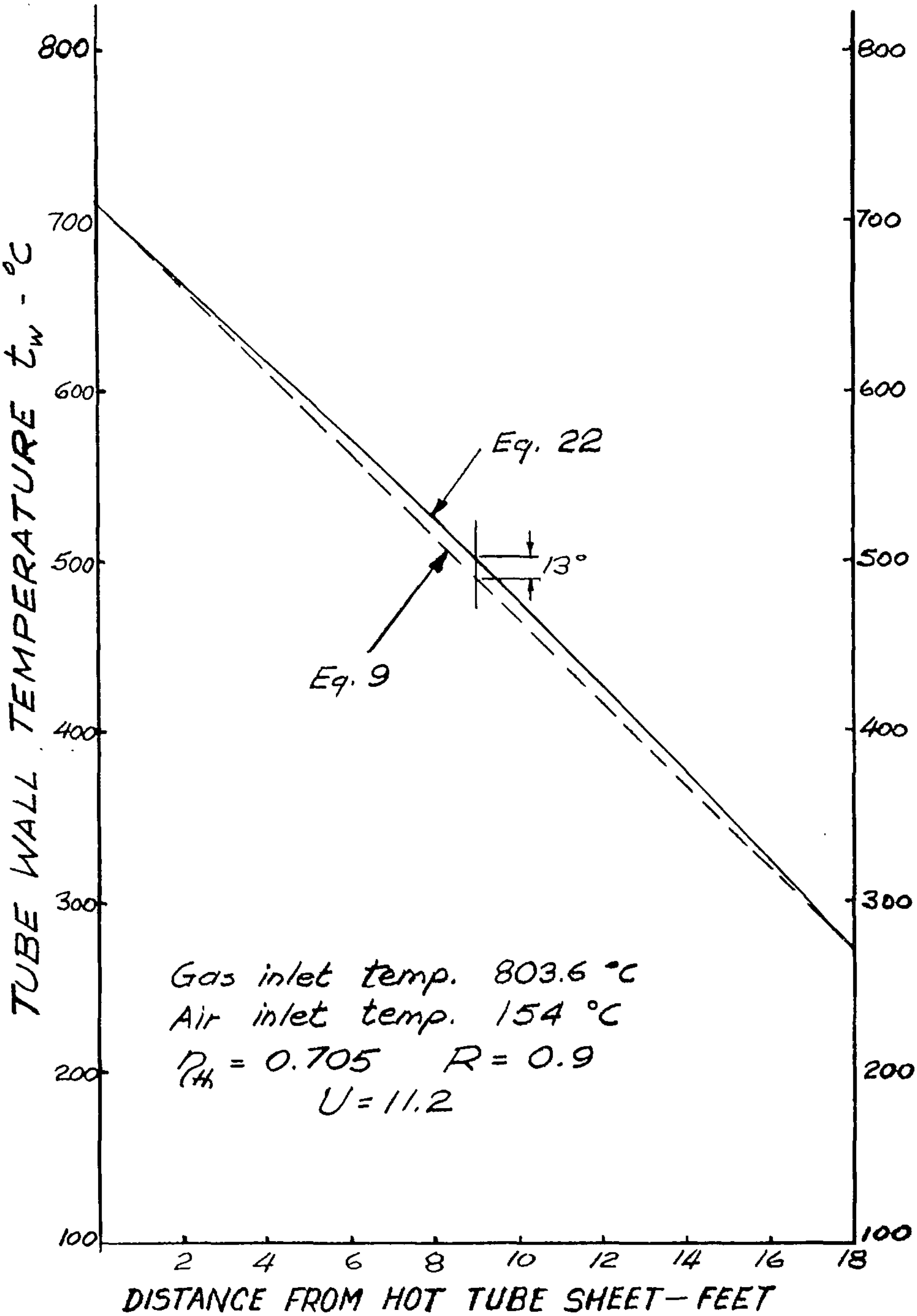

Figure A12-1. Tube-wall temperatures.