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The Inverted Brayton Cycle for Waste-Heat Utilization[1]

D. G. WILSON
Professor,
Mechanical Engineering,
MIT,
Cambridge, Mass.
Mem. ASME

N. R. DUNTEMAN
Project Engineer,
Electromotive Div.,
General Motors Corp.,
Detroit, Mich.

The inverted Brayton cycle, which can be simply defined as one in which hot gas is first expanded through a turbine to low pressure, is then cooled at constant pressure, and lastly, is recompressed to the initial pressure, has been shown to give attractive incremental gains in thermal efficiency, and large returns on investment, when added to a conventional shaft-power gas turbine exhausting into a waste-heat boiler. When the inverted Brayton cycle is applied by itself as a method of obtaining shaft power from the hot waste gas stream, there appears to be range of temperature and pressure ratios at which the cycle is competitive with other methods of waste-heat utilization.

[1]This paper is based on work carried out for Mr. Dunteman's M.S. thesis.

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, UNITED ENGINEERING CENTER, 345 EAST 47th STREET, NEW YORK, N.Y. 10017

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D. G. WILSON

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INTRODUCTION

The ideal Brayton cycle involves an enclosed working fluid being taken through a succession of processes consisting of isentropic compression, constant-pressure heat addition, isentropic expansion, and constant-pressure heat rejection, Fig. 1(a). Although there are some closed-cycle gas turbines which operate on an approximation to this cycle, most gas turbines use a so-called "open cycle" in which the atmosphere substitutes for the constant-pressure cooling process, Fig. 1(b). (The atmosphere also regenerates the exhaust gases so that internal combustion can be used.)

The closed cycle can be "opened" to the atmosphere at any point. In the usual open cycle, the turbine exhausts to, and the compressor draws from, the atmosphere. The inverted cycle is simply the name given to the gas turbine process in which the pressure (but not necessarily the temperature) at inlet to the turbine expander is approximately atmospheric. The inverted Brayton cycle as an independent prime mover consists of atmospheric air being drawn through a source of heat, such as a combustor; being expanded from high temperature and approximately atmospheric pressure to subatmospheric pressure; being cooled at approximately constant, low, pressure; and finally being recompressed to atmospheric pressure and being discharged (Fig. 2).

The thermal efficiency and specific power output of inverted cycles were evaluated as functions of temperature and pressure ratios by Hodge.[2] So far as we have been able to determine, no inverted-cycle gas turbine on this principle has ever been built and run. The reasons are obvious. All the components, and particularly the turbine and compressor, are much larger than those of a gas turbine working on the direct open cycle with high-pressure combustion. Expressed another way, a given compressor-turbine combination will produce many times the power operating with compression from atmospheric pressure than can be achieved when expanding from atmospheric pressure. Moreover, although the thermal efficiency for the given combination of temperature and pressure ratio should be identical in the two cases, the inverted cycle adds a component, the cooler, which is not necessary in the simple, direct, cycle and which adds a penalizing pressure drop; therefore,

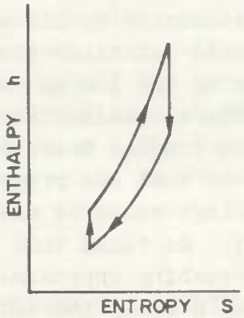

Fig. 1(a) Ideal Brayton cycle

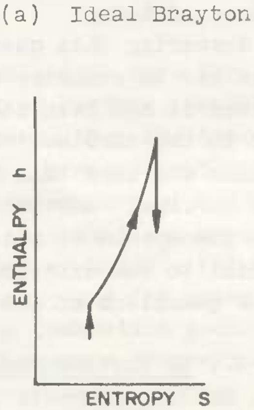

Fig. 1(b) Ideal open cycle

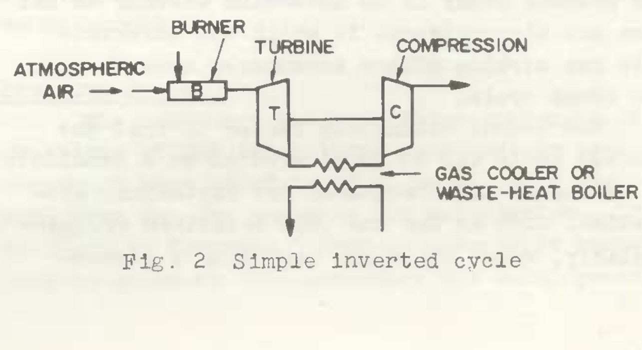

Fig. 2 Simple inverted cycle

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[2] Hodge, James, Gas Turbine Cycles and Performance Estimation, Butterworths, London, 1955.

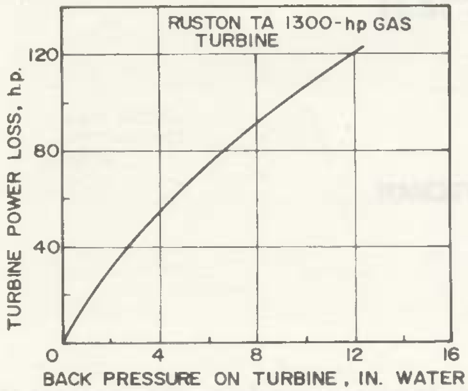

Fig. 3 Effect of back pressure on turbine power output

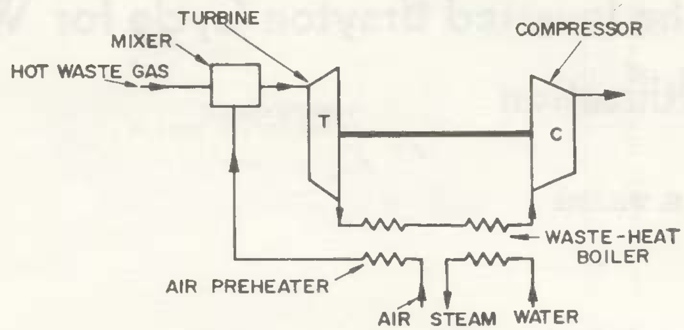

Fig. 5 Generalized inverted cycle

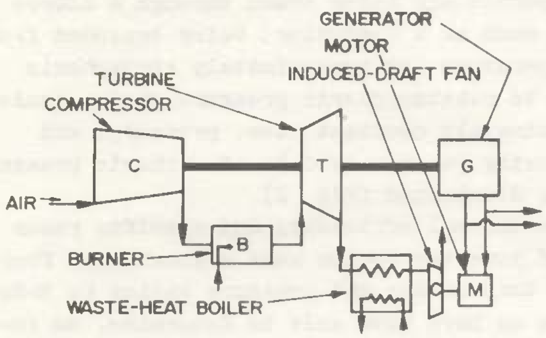

Fig. 4 Use of induced-draft fan with waste-heat boiler

the inverted-cycle thermal efficiency is lower.

The unattractiveness of the inverted cycle would seem to have been firmly established, and there would seem to be no reason to give it any further consideration.

There are, however, two mitigating factors of some importance. First, the inverted gas-turbine should be considered for use only when the heat input is free. In other words, it can be considered when there is a flow of gas at atmospheric pressure which is going to waste, or which might be incurring a thermal-pollution penalty. A conventional way of using waste heat in this form is to lead the gas to a waste-heat boiler and, if power is desired, to incorporate this boiler in a steam-power cycle. The purpose of this present study is to determine whether or not there are circumstances in which the inverted-cycle gas turbine offers advantages over the use of a steam cycle.

The second mitigating factor is that the inverted cycle can be considered to be a candidate for adding to an arrangement for waste-heat utilization, such as the one just described or, particularly, where the waste heat from a conventional gas turbine is utilized. The attractive potential of this form of the inverted cycle was brought to light in the following way.

Many conventional gas turbines are sold in a so-called "total-energy" package in which a waste-heat boiler is incorporated to remove sensible heat from the turbine exhaust.

A performance penalty must normally be accepted when one adds an exhaust-heated boiler to a gas turbine. The penalty is usually given in the form shown in Fig. 3, which is for an early version of the Ruston and Hornsby TA 1250 kw gas turbine.[3] The addition of a pressure drop to the turbine exhaust produces power-output reduction which is linear in the region of interest.

A Canadian purchaser of a TA turbine wrote to Ruston and Hornsby to inform them that their performance penalty seemed to be in error. The reason for his statement was that he had decided to add a waste-heat boiler to his existing TA turbine, but he could not allow the turbine power output to decrease by the 100 hp which was predicted. He, therefore, decided to install an electrically driven induced draft fan after the waste-heat boiler so that the pressure at the turbine-exhaust flange would be restored to atmospheric (Fig. 4). He found that he could do this with a fan absorbing approximately 60 hp. How could he obtain a power increase of 100 hp with the expenditure of 60 hp?

One way of answering this question is to state that the fan had to compress a smaller volume of gas, since it had been cooled, than had been expanded in the turbine. Therefore, its power absorption was less than the restored power gain of the turbine. Another way of describing the same phenomenon is that an inverted cycle had been added to the existing gas turbine.

Some further questions of interest imme-

[3] Forbes, S. M., TA Turbine Performance Data, T. D. report 125, Ruston & Hornsby Ltd., Lincoln, England, Nov. 1956.

Table 1 Comparative Results of Cycle-Calculation Methods

Method	Thermal efficiency, percent	Specific power, hp/lbm/sec
Constant-specific-heat methods		
Method of Hodge[2]	7.0	28.0
True mean specific heat	7.33	27.2
Gas-table methods		
Pure air	7.59	27.62
400% theoretical air	7.63	27.90

diately arise. If restoring the turbine-exhaust pressure to atmospheric produces a net power gain over the case where the boiler has no induced-draft fan, does decreasing the turbine-exhaust pressure to below atmospheric produce a greater gain? Assuming that it does (as will be shown later), by how much must the waste-heat boiler be increased in size to meet the same design-point conditions? Can the added induced-draft fan pay for itself in different circumstances?

These questions, and similar questions addressed to the case of the inverted cycle operating independently of a gas turbine, are examined in this paper. The results of economic optimizations show that, in certain circumstances, the inverted cycle shows very attractive rates of return (Fig. 5).

THERMODYNAMIC PERFORMANCE STUDIES

Specific Heat

Hand calculations were first made to determine the effect of various simplifications to the cycle analyses. For the test case, a cycle was chosen with a pressure ratio of 3.42 and a hot-gas inlet temperature of 1570 Rankine (873.2 K) was chosen. Calculations were made based: (a) on the method of Hodge[2]; (b) using a true mean specific heat; and (c) using tables for pure air and for combustion products of a hydrocarbon with 400 percent theoretical air.[4] The

Table 2 Values used in Cycle Calculations

Pressure drop	7.5 percent
Leakage	variable
Turbine polytropic efficiency	85 percent
Compressor polytropic efficiency	87 percent
Heat-exchanger effectiveness	90 percent
Ambient temperature	27 C

results of these calculations are summarized in Table 1.

The substantial discrepancy in the method of Hodge occurred because of the assumption that the compressor pumps air, which is not the case for this inverted cycle. With the correct value of specific heat for the compressor, good agreement with the method of true mean specific heats was found.

For this cycle, the gas properties should have been those of 500 percent theoretical air so that some interpolation between the results for 400 percent theoretical air and for pure air should be made for maximum accuracy. However, the results are close for these two cases.

For the computer calculations, the method of true mean specific heats was used for simplicity, with apparent errors of up to 0.3 percentage points, or 4 percent, in thermal efficiency, and 2.5 percent in specific power. An additional simplification in the computer calculations was that pure air properties with mean specific heats for mean temperatures, together with the polytropic exponent associated with that mean temperature, were used. In practice, a wide variety of different gas compositions might be expected, so that recalculation on an exact basis using the present results as a guide would normally be correct procedure.

Combustion Efficiency

The combustion efficiency was assumed to be 100 percent in cycles where fuel was assumed to be burned.

Pressure Drops

The component pressure drops expressed as fractions of the inlet total pressures to each component were added together as a single pressure drop for the purposes for calculation. It is shown is footnote [2] that an acceptable accuracy is given by this procedure for total pressure

[4] Keenan, J. H., and Kaye, Joseph, Gas Tables, Wiley, New York, 1948.

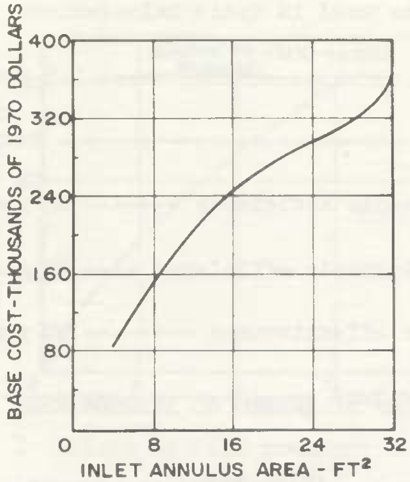

Fig. 6(a) Base cost of turbine expanders

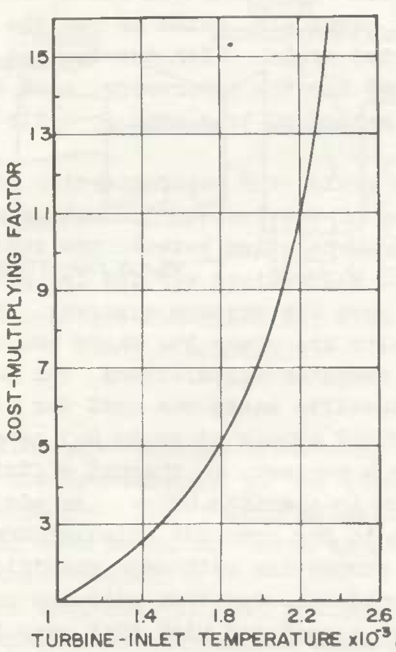

Fig. 6(b) Effect of inlet temperature on turbine costs

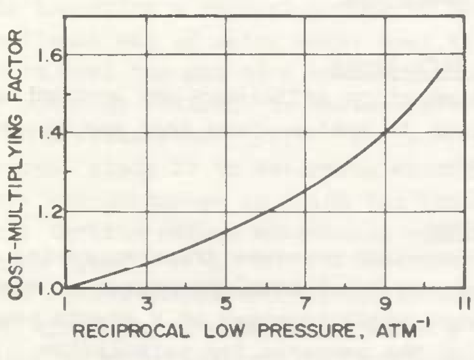

Fig. 6(c) Effect of vacuum conditions on turbine and compressor cost

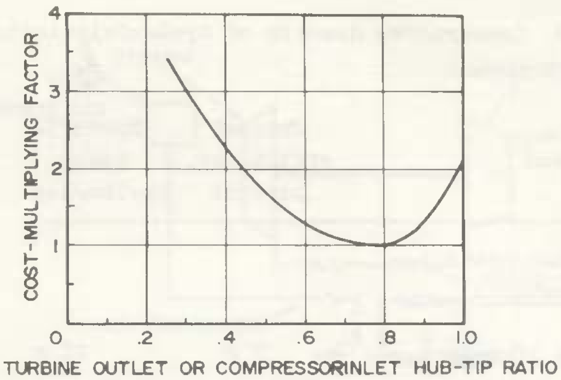

Fig. 6(d) Effect of low-pressure-end hub-tip ratio on compressor and turbine cost

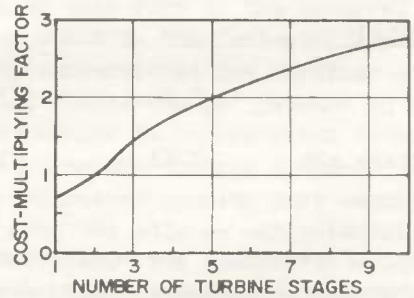

Fig. 6(e) Effect of number of stages on compressor and turbine cost

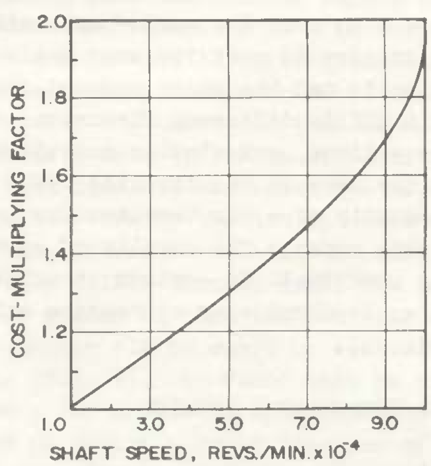

Fig. 6(f) Effect of shaft speed on compressor and turbine cost

drops of up to 25 percent, which is substantially higher than was used in any of the present cases. A standard value of 7.5 percent was employed.

Component Efficiencies

Polytropic total-to-static efficiencies were used for the compressor and turbine. Conservatively low values of 85 percent for the turbine and 87 percent for the compressor were em-

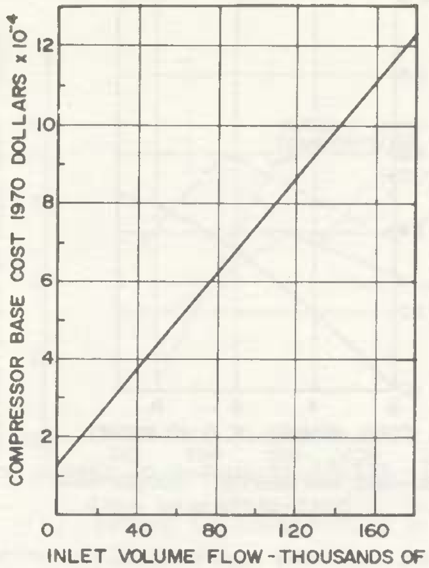

Fig. 7(a) Base cost of compressors

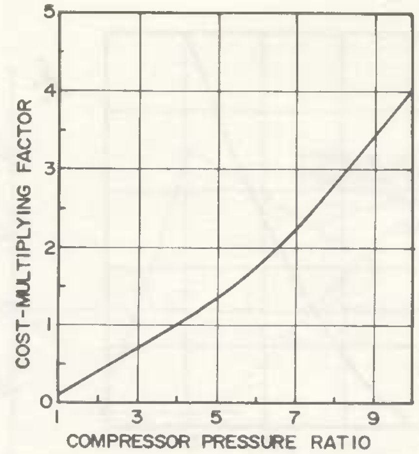

Fig. 7(b) Effect of pressure ratio on compressor cost

ployed. Industrial turbines and compressors are running with polytropic efficiencies of well over 90 percent, but the present low values were retained to incorporate accessory power losses and other incidentals which were not evaluated separately.

A leakage coefficient, which was a function of blade length and diameter, was used.

Heat-Exchanger Effectiveness

An overall effectiveness of 90 percent was used for all cases.

Ambient Conditions

The ambient conditions were assumed to be 1-atm pressure with a temperature of 27 Celsius (300 K; 540 R).

COST STUDIES

Because a fundamental assumption about the value of using an inverted cycle is that the hot-gas inlet for this cycle would be free (in other words, it would be a normally waste gas stream), the only reasonable criterion which can be used to evaluate the usefulness of this cycle is the net sum of the cost incurred and the benefits to be realized.

Costs are notoriously difficult to estimate. It is easy to find examples where two experienced cost engineers have produced estimates which differ one from the other by a factor of ten, even for established and readily available engineering components. When one enters the realm of new forms of machinery, the ranges can be even wider.

Our cost estimates, therefore, had to be crude and are open to question. We believe them to be conservative in most respects and to give useful guidance as a starting point to more thorough evaluation of the economic viability of these arrangements. As will be shown, our most favorable arrangement showed a return on investment of 34 percent, which would presumably mark it as worthy of further study however skeptical one was about the basis for the estimate.

Capital costs of the turbine, heat exchangers, and compressor in each cycle were estimated. The value of the power sent out was considered to be the gross income for the plant. The present net worth was then evaluated simply as the present value of the power produced less the total capital costs.

The average return on investment was also calculated as the value of the power produced for one year less the depreciation of the total capital costs, all divided by the capital costs.

The capital costs were assumed to include the costs of foundations and buildings and the maintenance costs for the life of the plant. We did not include the costs of supervision. We assumed that these plants would run either with remote supervision as do standby power plants and industrial units of many types, or would require no additional supervision when added to an existing plant.

Capital costs were estimated by using a known cost for an average-size and average-duty component and by then estimating the variation of cost with size and duty. Fig. 6(a) shows the assumed cost curve for single-stage, moderate-temperature axial turbines, expressed as a function of the inlet-annulus area. Fig. 6(b) shows

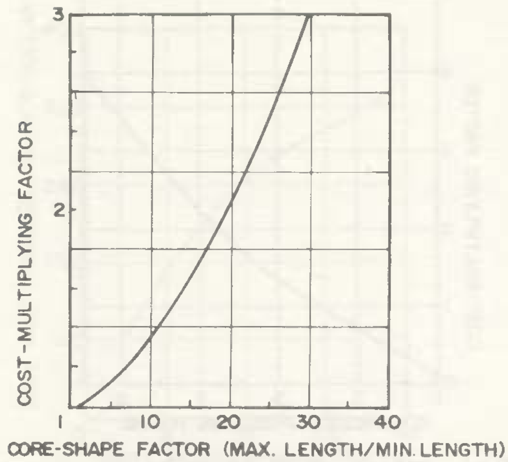

Fig. 8(a) Effect of core shape on heat exchanger cost

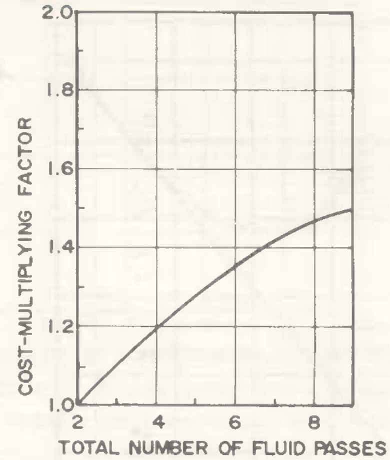

Fig. 8(c) Effect of number of passes on heat-exchanger cost

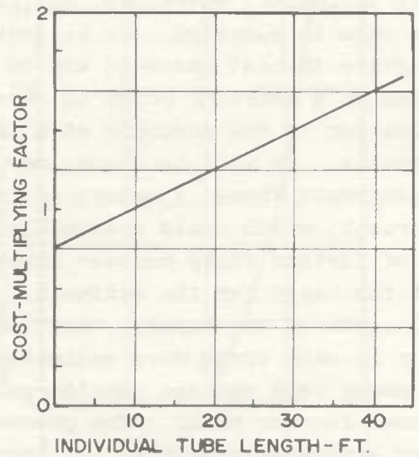

Fig. 8(b) Effect of tube or core length on heat-exchanger cost

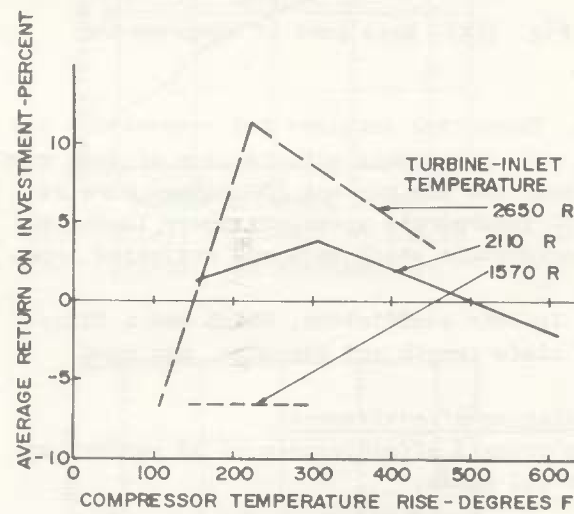

Fig. 9(a) Return on investment for generalized inverted cycle (Power: 5000 hp, gas inlet temperature: 3500 R)

an assumed cost-multiplying factor for higher turbine-inlet temperatures. Fig. 6(c) shows assumed cost-multiplying factors for the effects of constructing a turbine to withstand a vacuum. For given annulus areas, Fig. 6(d) shows the increase in costs which was assumed to result from using a hub-tip ratio at turbine outlet (or compressor inlet) which differs from 0.75. Fig. 6(e) covers the multiplying factor for the number of turbine stages, and Fig. 6(f) shows an assumed curve for increases in cost resulting from shafts speeds above 10,000 rpm.

The assumed base costs for axial compressors are shown in Fig. 7(a) with the effects of pressure ratio indicated in Fig. 7(b). Other multiplying factors for the costs of axial compressors, such as the effects of vacuum, hub-tip ratio, and rpm are assumed to be the same as for turbines.

Heat Exchangers

The heat exchangers are assumed to be plain-tube multi-pass cross-counter-flow units. The base heat exchanger is assumed to have 1/4-in. tubes in a close staggered array, and a base cost of $500/cu ft was assumed. A heat exchanger in the form of a cube is assumed to have a minimum cost at this value, and heat exchangers of other sizes and shapes are assumed to be more expensive according to the cost-multiplying factors shown in Fig. 8(a).

Cost-multiplying factors for tube length and for number of passes are shown in Figs. 8(b) and 8(c).

Value of Mechanical Energy

It is assumed that the shaft power would

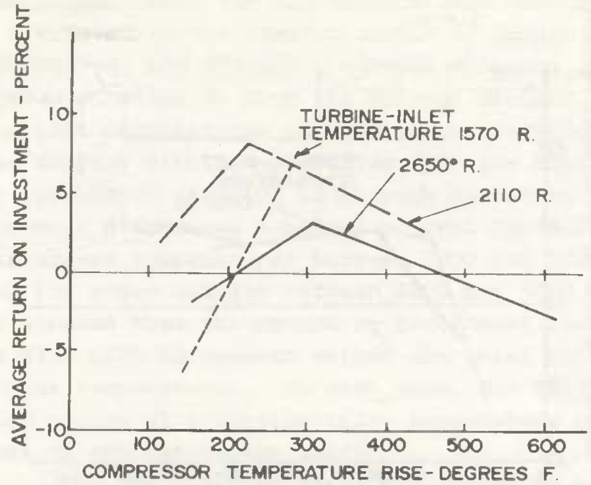

Fig. 9(b) Return on investment for generalized inverted cycle (Power: 2000 hp, gas-inlet temperature: 3500 R)

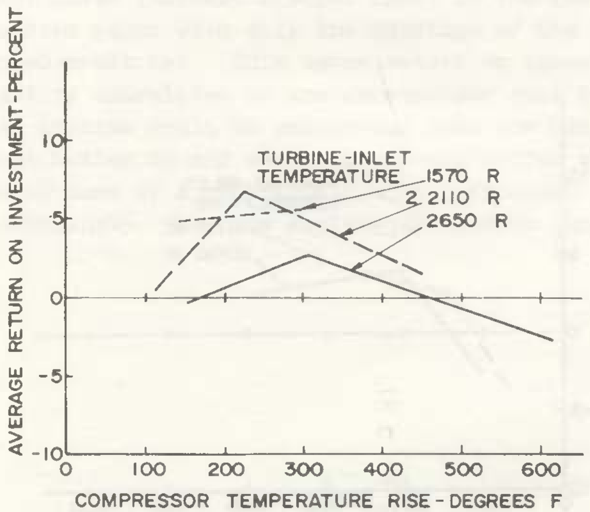

Fig. 9(c) Return on investment for generalized inverted cycle (Power: 1000 hp, gas-inlet temperature, 3500 R)

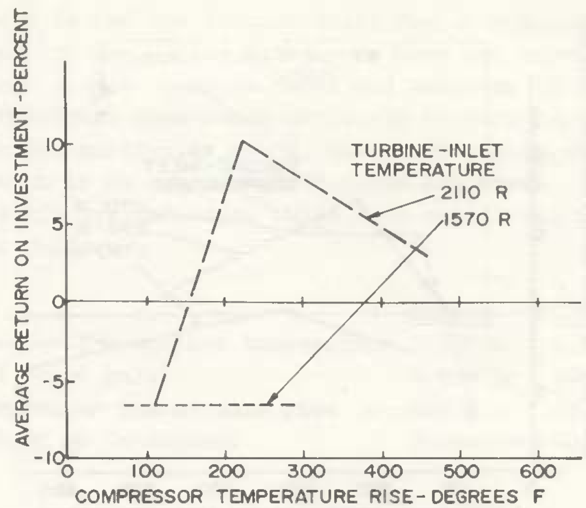

Fig. 9(d) Return on investment for generalized inverted cycle (Power: 5000 hp, gas-inlet temperature, 2750 R)

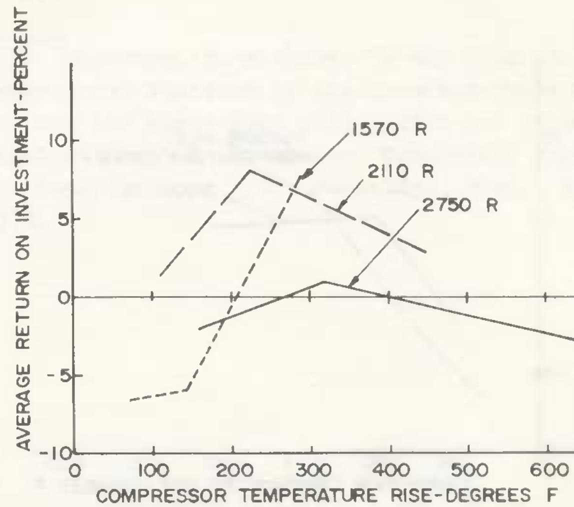

Fig. 9(e) Return on investment for generalized inverter cycle (Power: 2000 hp, gas-inlet temperature, 2750 R)

be used for purposes which would normally require an electric motor, and, accordingly, the value of power is taken on the low side of the average of industrial electric-power rates at \$0.005/kw-hr. A continuous duty cycle is assumed.

Capital-Cost Depreciation

A straight-line depreciation rate is assumed based on a life time of 15 years.

Interest Rate

An interest rate of 8 percent is used. When coupled with the assumed life time of 15 years, the present net worth of the mechanical power produced during 15 years becomes 8.53 times the value of the power during one year.

Profitability

The profitability is defined simply as the following.

Profitability ≡
present value of power produced - capital costs.

Return on Investment

The average return on investment (ROI) is defined as follows.

ROI ≡

$$\frac{\text{value of power for one year - depreciation for one year}}{\text{capital costs of plant}}$$

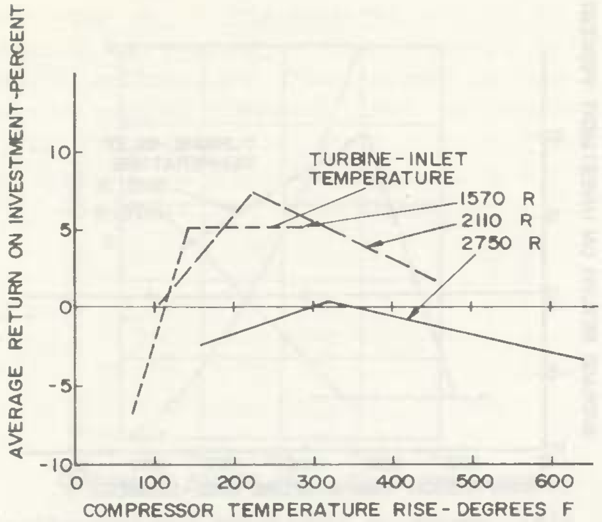

Fig. 9(f) Return on investment for generalized inverted cycle (Power: 1000 hp, gas-inlet temperature, 2750 R)

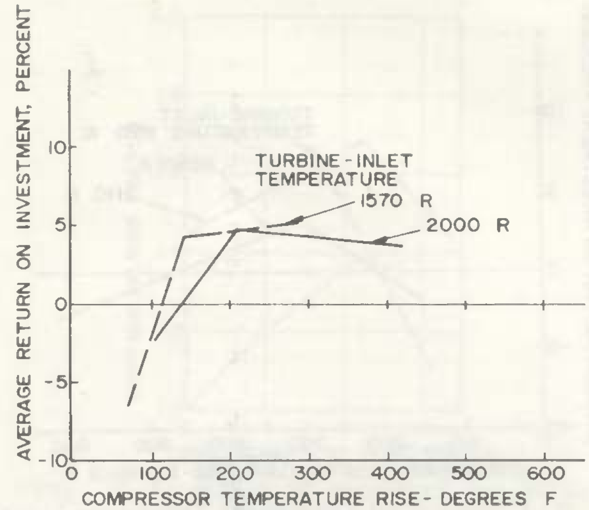

Fig. 9(h) Return on investment for generalized inverted cycle (Power: 1000 hp, gas-inlet temperature, 2000 R)

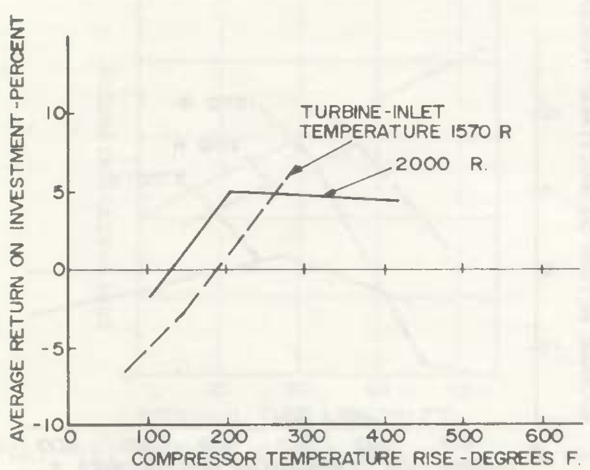

Fig. 9(g) Return on investment for generalized inverted cycle (Power: 2000 hp, gas-inlet temperature, 2000 R)

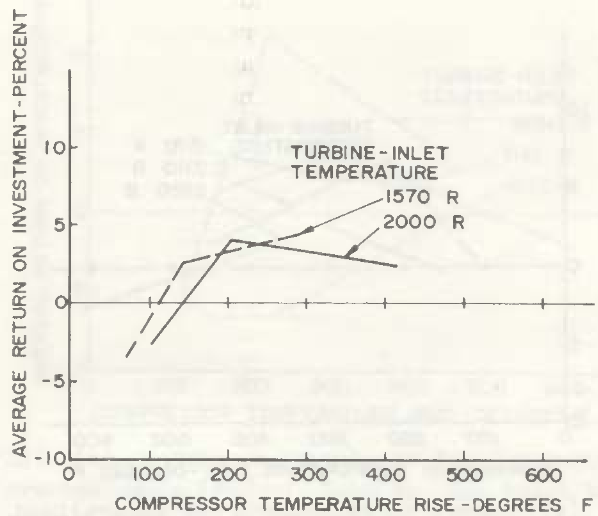

Fig. 9(i) Return on investment for generalized inverted cycle (Power: 500 hp gas-inlet temperature, 2000 R)

Add-On to Total-Energy Plant

The costs for the conversion of a standard total-energy plant, consisting of a gas turbine exhausting into a waste-heat boiler, are considered to be simply the cost of having a compressor downstream of the boiler. The boiler would then operate under vacuum instead of under slight positive pressure, and the turbine would have an increased, rather than a decreased, pressure drop. It is assumed that no changes would be required to the turbine or boiler design.

Results

The results are shown in Fig. 9. Before the specific results are discussed, some general comments are made on the economic comparisons.

The pressure ratios for all cycles, at which most favorable economics were found, were at pressure ratios nearer unity than to those values giving optimum thermodynamic efficiency. Smaller-than-optimum pressure ratios lead to higher pressures for the vacuum part of the cycle and, therefore, to smaller components.

With the cost relations used here, the compressor and turbine costs were found to greatly outweigh the costs of the heat exchangers in all the independent inverted cycle of interest. This was not the case, however, with the inverted cycle as an add-on device for a gas turbine, when no additional costs for the turbine were assumed.

Based on the limited number of design points considered, and without a careful choosing of pressure ratios to find the optimum designs. the cost calculations for the inverted-cycle gas turbine with hot-gas inlet indicate that it is capable of yielding an average return on investment of between 6 and 10 percent for inlet gas-stream temperatures between 2000 and 3000 F and for power outputs between 1000 and 5000 hp. It appears that the return on investment increases with both horsepower output and inlet gas-stream temperatures. In each case, the best design occurs at a turbine-inlet temperature below that of the gas-stream inlet.

For the inverted cycle considered as an add-on device (waste-heat boiler and induced-draft fan) for a standard power-producing gas turbine, the return on investment seems to be of the order of 30 percent because of the significant power increase brought about in the gas-turbine plant with only the addition of the induced-draft fan. This large return on investment is calculated on the assumptions that the gas turbine would be exhausting into the waste-heat boiler in any event, and would suffer a power loss as a result of the back pressure. Accordingly, the only capital investment necessary is for the induced-draft fan or compressor and its associated drive. We have not calculated cost curves, because costs and benefits in each individual case would obviously be very dependent on the particular plant, the primary duty of which is to produce shaft power and steam. Two fairly typical cases which were calculated were as follows.

	Case 1	Case 2
Gas-turbine exhaust temperature	1,500 R	1,200 R
Net Power gain	1,000 hp	500 hp
Compressor temperature rise	264 R	173 R
Return on investment	34 percent	21.9 percen

ACKNOWLEDGMENTS

During the period of this study, Mr. Dunteman was supported at M.I.T. by a General Motors fellowship. This paper is based on his masters thesis.[5]

5 Dunteman, N. Richard, "A New Look at the Competitive Position of the Inverted-Cycle Gas Turbine for Waste-Heat Utilization and Other Applications," M. S. thesis, Mechanical Engineering Department, M.I.T., Cambridge, Mass., Aug. 1970.