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Off-Design Analysis of a Gas Turbine Powerplant Augmented by Steam Injection Using Various Fuels

Robert J. Stochl
Lewis Research Center
Cleveland, Ohio

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OFF DESIGN ANALYSIS OF A GAS TURBINE POWER PLANT AUGMENTED BY STEAM INJECTION USING VARIOUS FUEL TYPES

by Robert J. Stochl

National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

SUMMARY

The results of an analysis to estimate the off-design performance of a specific gas turbine powerplant with steam injection are presented. Results were obtained using coal-derived low and intermediate heating value fuel gases and a conventional distillate. The turbomachinery characteristics used in the calculation of off-design operation were scaled from available performance maps of aircraft type gas turbines.

The results indicate that steam injection does provide substantial increases in both power and efficiency within the available compressor surge margin. The results also indicate that the increase in performance with steam injection is insensitive to the type of fuel. Also, in a cogeneration application, steam injection could provide some degree of flexibility by varing the split between power and process steam.

INTRODUCTION

A concept for simultaneously increasing the power output and efficiency of a gas turbine is based on the injection of steam into the combustor of the gas turbine. Steam injection increases the power output of a gas turbine by incresing the mass flow (and its specific heat) through the turbine without significantly increasing the power required to drive the compressor. If the steam is generated by recovering otherwise wasted heat from the gas turbine exhaust, the power system efficiency would also be increased. From this standpoint a steam-injected gas turbine cycle is thermodynamically similar to a combined gas-turbine - steam-turbine cycle. However, the steam portion of the steam injected cycle operates at much lower pressure levels, and higher temperature levels than the steam portion of a conventional combined cycle. It also has a potential cost advantage of not requiring a separate steam turbine generator or its heat rejection system.

Several investigators (refs. 1 to 4) have analytically shown that a considerable increase in performance (both power and efficiency) can be achieved by injecting steam into the combustor of a gas turbine. Although the authors of reference 2 did consider a variation in compressor-pressure ratio with increased mass flow, most analyses have estimated the performance of a gas turbine at each steam injection rate assuming design-point performance. They have not included the analysis of a fixed set of hardware for a variation in the amount of injected steam. A potential advantage of steam-injection gas turbines is the ability to change power level by changing the steam-injection rate, while holding the turbine-inlet temperature at a constant value. This could result in attractive performance over a range of power outputs for a fixed machine.

One purpose of this investigation was to determine the effect of steam injection on the performance of a gas-turbine system using turbomachinery not specifically designed for steam injection. The system operating parameters (such as turbine-inlet temperature and compressor-pressure ratio), used in this report were based on presently available, advanced, single-shaft, heavy-duty gas turbines. The turbomachinery characteristics used in the calculation of off-design operation were scaled from available performance maps of aircraft-type gas turbines. These maps would not necessarily match those of heavy-duty industrial gas turbines but should be suitable to indicate the proper trends and to assess the potential performance advantages in using steam injection in gas turbines.

When the amount of steam injection is increased in a particular fixed-geometry gas turbine, the turbine flow increases relative to the compressor flow, increasing the compressor back pressure and moving the compressor pressure ratio toward the surge line. The initial surge margin available to the compressor depends on its design operating point. Because the operating point of a particular gas turbine might be different when used with a coal-derived gaseous fuel compared with a conventional liquid fuel, it is of interest to consider a variety of fuel types in assessing the advantages of steam injection. Also, in a power system integrated with a gasifier, it might be necessary to start up using a fuel other than that produced by the gasifier or to be able to operate on a second fuel if the gasifier for some reason must be shut down. Therefore, the performance of a specific gas turbine with steam injection was considered using coal derived low and intermediate heating value fuel gases, as well as a conventional distillate. Systems using low heating value fuel from both an atmospheric gasifier and from a pressurized gasifier were considered. The emphasis here is not to evaluate or compare gasifier designs but to determine the effect of the fuels obtained from the various gasifiers. Calculations were performed assuming that the turbomachinery operated at the design point for only one of these fuels. The operating point was then calculated for each of the other fuels. In each case a range of steam injection rates were considered. Two sets of such calculations were made: one assuming that the turbomachinery operated at its design point when low heating value fuel from an atmospheric gasifier is used, the other assuming that intermediate heating value fuel from a pressurized gasifier is used.

The turbine-inlet temperature was assumed constant at 1093° C (2000° F). The effects of turbine cooling were not included in the analysis. The inlet steam conditions to the combustor were assumed to be 482° C at 1.379 MPascal (900° F at 200 psia) in all cases.

DESCRIPTION OF SYSTEMS AND ASSUMPTIONS

Four basic gas turbine system configurations were evaluated. The basic distinction between the four configurations is the type of fuel used.

Configuration 1 - Simple steam-injected gas turbine fueled by the products of an air-blown coal gasifier operating at atmospheric pressure.

Configuration 2 - Simple steam-injected gas turbine fueled by the products of an air-blown pressurized coal gasifier.

Configuration 3 - Simple steam-injected gas turbine fueled by the products of an oxygen-blown pressurized coal gasifier.

Configuration 4 - Simple steam-injected gas turbine fueled by a light distillate.

Configuration 1 is shown schematically in figure 1(a). Ambient air at 16° C (60° F) and 0.010 MPascals (14.7 psia)) is pressurized in a compressor, heated in the combustor together with injected steam by the burning of fuel from the gasifier, and expanded in the turbine. The turbine drives the compressor and generator. The heat in the turbine exhaust is used to raise 482° C (900° F), 1.379 MPascal (200-psia) steam in a heat recovery boiler, a portion of which is used for steam injection (the remainder could be used as process steam). The turbine-inlet temperature was maintained at 1093° C (2000° F) in all cases. The fuel from the gasifier was assumed to enter the gas turbine combustor at 316° C (600° F) and at a pressure 0.334 MPascals (48.5 psia) above the compressor exit pressure. The composition of the fuel going into the combustor is given in table I for each configuration. The auxiliary power requirement included in this configuration was that required to raise the low heating value fuel gas to a pressure 0.334 MPascals (48.5 psia) above the main compressor-discharge pressure.

Configuration 2 (fig. 1(b)) is similar to configuration 1 except that the pressurized air required by the gasifier is supplied from the gas-turbine compressor. The auxiliary power requirement included in this configuration is the power necessary to increase the air supply pressure by 0.334 MPascals (48.5 psia) in a boost compressor. The gasifier air input to fuel output ratio (Ma_2/Mf - 0.7628) and fuel gas composition as shown in table I were assumed constant and unaffected by gasifier pressure level.

Configuration 3 (fig. 1(c)) is a simple gas turbine fueled from an oxygen-blown pressurized gasifier. The oxygen is produced from air and pressurized to the operating pressure level of the gasifier (P + 0.334 MPascals (48.5 psia)). The gasifier was assumed to require 0.291 kilograms of oxygen per kilogram intermediate heating value fuel output. The fuel composition of the intermediate heating value fuel into the gas turbine combustion is given in table I. The auxiliary power requirements for this configuration were assumed to be (1) 0.255 kilowatt-hour ker kilogram of atmospheric pressure oxygen produced in the oxygen plant and (2) the power required to pressurize the oxygen to the gasifier operating pressure.

Configuration 4 (fig. 1(d)) is a simple steam injected gas turbine using a light distillate fuel. Fuel system auxiliary power would be relatively small and was not included.

It was assumed for all cases that the gasifiers were capable of operating, without change in efficiency, over the range of fuel flow rates required by the gas turbine for the various steam injection rates. The lower stack gas temperature limit for the turbine exhaust was assumed to be 149° C (300° F). This limit, as will be discussed later, in all but two cases restricted the maximum amount of 482° C (900° F) steam that could be produced by recovering heat from the turbine exhaust.

With the exception of configuration 4, each of the fuel supply systems are in various degrees integrated with the power conversion system. It is not the purpose of this report to evaluate or compare the various gasifier systems. However, to properly compare system performance for the various fuels some consideration of gasifier performance was required. Included in the net work output of the gas turbine systems are the auxiliary power requirements needed to supply the fuel gas to the gas turbine combustor at a pressure of 0.334 MPascals (48.5 psia) above combustor pressure. Auxiliaries associated with coal preparation or fuel gas clean up were not included. Figure 2 is an energy flow diagram for each of the three coal derived fuel systems. These data are the result of a heat balance of a fluid bed gasifier with cold gas clean up. In figure 2(b) and (c) the fuel gas is reheated by recuperation to 316° C (600° F). In each gasifier steam is produced in the process of cooling the fuel gas. Any part of this steam could be used for steam injection into the gas turbine combustor.

To arrive at the system efficiency the gas turbine cycle efficiency was multiplied by the effective efficiency, η_g, of the fuel supply systems. The η_g is equal to the fuel energy input to the gas turbine ($Q_{\eta,GT}$) divided by the coal energy input to the fuel supply system ($Q_{\eta,coal}$) and $Q_{\eta,GT}$ is based on only the higher heating value of the fuel gas and $Q_{\eta,coal}$ is based on the higher heating value of Illinois number 6 coal 28.46 Mj/Kg (12 235 Btu/lb). For configurations 1 and 2 $\eta_g = 0.760$, for configuration 3 $\eta_g = 0.806$.

As mentioned in the introduction, two sets of calculations were made. One, designated design option 1, is where system configuration 1 (without steam injection) operated with the parameters listed in table II. Configurations 2 to 4, with and without steam injection, as well as configuration 1, with steam injection, were then considered as operating in an off-design mode. The second set of calculations, design option 2, is where configuration 3 (without steam injection) operated with the parameters listed in table II. Here, configuration 1, 2, and 4, with and without steam injection, as well as configuration 3, with steam injection, were operated in an off-design mode. All calculations were performed using the compressor and turbine characteristics shown in figures 3. These characteristics, obtained by scaling a large aircraft-type turbomachine to an industrial size operating with the chosen design parameters, apply to a single-shaft gas turbine operating at its design speed. The pressure losses for each component with the addition of steam injection and the other fuels were assumed to change according to the relation

$$\frac{\Delta P}{P} = \left(\frac{\dot{M}}{\dot{M}_d}\right)^{1.75} \left(\frac{\sqrt{T}}{\sqrt{T_d}}\ \frac{Pd}{P}\right)^2 \left(\frac{\Delta P}{P}\right)_d$$

where

$\dot{M}$ is the mass flow rate,
T is the inlet temperature,
P is the inlet pressure,
ΔP is the component pressure drop,
and the subscript d indicates design condition.

DESIGN OPTION 1 RESULTS

The effect of fuel type on the performance of design option 1 turbomachinery is shown in table III. As stated previously, design option 1 turbomachinery was sized for the parameters listed in column 1 (configuration 1). The results shown are without steam injection (M_s/M_a = 0). There is a 28 to 35 percent decrease in the gross output for configurations 2 to 4 compared with configuration 1. The design fuel to air ratio for configuration 1 was 0.257. This means there is 25.7 percent more mass flow through the turbine than through the compressor, resulting in a higher gross work output per pound of airflow. As the heating value of the fuel increases (for intermediate Btu and light distillate fuels), the fuel to air ratio required to maintain 1093° C (2000° F) turbine inlet temperature decreases (i.e., $(F/A)_{Conf\ 3}$ = 0.076 and $(F/A)_{Conf\ 4}$ = 0.022) which lowers the gross specific work output. However, when the auxiliary power required to compress the fuel and/or oxidizer are deducted from the generator output, the resulting difference in net power between configuration 1 and the other three is reduced 12 to 15 percent. It should be noted that the relatively small amount of auxiliary losses shown for configuration 2 results because a major portion of the power required for the pressurization of the oxidizer is supplied by the gas turbine compressor and is, therefore, included in the determination of the gas turbine generator output.

There are three efficiency values shown in table III. Gas turbine efficiency (η_{GT}), which is defined as the generator output divided by the energy input to the gas turbine combustor based on the higher heating value of the fuel gas. Cycle efficiency (η_{Cycle}) is the generator output minus the auxiliary power required for motor driven air or fuel gas compressors (net power output), divided by the energy input to the gas turbine combustor. And system efficiency which is, except for configuration 4, the net power output divided by the coal energy input to the gasifier based on the higher heating value of the coal. For configuration 4 the light distillate fuel was assumed not be derived from coal and to require no auxiliary power so the three efficiency values are the same. The gas turbine efficiency is also highest for configuration 1. This is to be expected since the turbomachinery in the other configurations would be operating at an off-design point with other compressor efficiencies and lower pressure ratios. Once the auxiliary losses and gasifier efficiencies are included, the cycle and system efficiencies are higher for the intermediate heating value and light distillate fuels (configurations 3 and 4). The detailed performance results using steam injection are tabulated in table IV. The performance results are presented in figures 4 to 6.

Figure 4 illustrates the effect of steam injection on the net power output for the four configurations. The trend for all configurations is an approximately linear increase in net power for increasing steam to air ratios. There is approximately an 8 electrical megawatt (MW_e) increase in power for each 0.1 increase in the steam to air ratio. The maximum amount of steam that can be injected into the combustor is limited by the surge margin of the compressor The compressor-pressure ratio increase toward the surge line as the steam-injection rate increases. The steam to air ratio at which compressor surge is encountered is indicated in figure 4. It varies from a steam to air ratio of slightly more than 0.2 for configuration 1 to 0.5 for configuration 4. In all configurations steam can be produced by recovering the heat in the turbine exhaust. In the configurations using the coal-derived fuel gases (configurations 1 to 3) additional steam is generated in the gasification process. (See fig. 2.) The limits on the steam to air ratios for these two steam-generating sources are also indicated in figure 4. For configuration 1 the compressor surge limit is reached before the steam generating capacities are exceeded. For configurations 2 to 4 the steam raising capability in the turbine exhaust limits the steam to air ratio to approximately 0.22. For configuration 2 this steam to air ratio can be increased to 0.37 by also using steam produced in the gasification process, while the steam to air ratio for configuration 3 can only be increased to 0.26 using gasification steam. The performance results that follow, for design option 1, will be restricted to the appropriate limiting steam air ratios.

Figures 5 and 6 present the cycle and system efficiencies over the appropriate steam to air ratios. The trend for all configurations is increasing cycle and system efficiencies for increased steam to air ratio. There is approximately a 32-percent increase in system efficiency (fig. 6) for configuration 1 for its limiting steam to air ratio. For configurations 2 to 4 there is approximately a 40 to 62 percent increase.

It should be noted that the system efficiency for the light distillate fuel (configuration 4) is considerably higher than for the coal derived fuels, because that power system was not charged with any fuel conversion efficiency or any auxiliary power losses connected with the supply of fuel. If the light distillate was derived from coal the system efficiency values shown would then have to be modified by a conversion efficiency factor which would bring them more in line with other coal derived fuels. If the light distillate was not derived from coal its cost of power would be higher than the coal derived fuels.

Figure 7 is a composite of figures 4 and 6, that illustrates the relation of system efficiency, net power output, and steam to air ratio. With steam injection the percent increase in efficiency is less than the percent increase in net power output due to the additional fuel required to heat the injected steam to the turbine-inlet temperature. In a cogeneration application the heat in the turbine exhaust might also be used to generate steam for process use as well as for injection. Also, the steam generated in the gasification process can be used for either injection or process. To follow a variation in power versus steam requirements, it might be desirable to vary the amount of steam injection versus the amount of steam used for process.

Figure 8 illustrates the relation between steam available for process use and the steam used for injection into the combustor. Two sets of results are shown. One is the result of using only the steam produced by reducing the turbine exhaust to the limiting stack temperature. The other is the result of using the steam produced from the turbine exhaust plus steam available from the particular gasifier. The maximum amount of 482° C (900° F), 1.379 MPascals (200-psia) steam that could be raised with and without gasifier steam is shown along the $M_s/M_a = 0$ coordinate. The amount of available process steam decreases linearly for increased injection ratios until all the steam raised is used for injection, or in the case of system configuration 1, the compressor surge limit is reached. The difference between the two sets of results is the steam produced in the gasification process. Both low heating value gasification subsystems produce considerably more steam than the intermediate heating value subsystem. The pounds of steam per pound of fuel gas output are slightly higher for the low heating value subsystem (0.458 and 0.368 for the atmospheric and pressurized low heating value subsystem, respectively, compared with 0.309 for the intermediate heating value system). However, the total fuel required using the low heating value fuel gas is more than three times that required of the intermediate heating value gas, resulting in much larger total steam production.

The relation between the total steam available (including any steam generated in the gasification process) for process use and net power is illustrated in figure 9 over the allowable range of steam to air ratios. This figure is a composite of figures 4 and 8. This information illustrates the potential ability of these systems to follow variation in power versus steam requirements. Both the net power and process steam production is greatest for system configuration 1 at any given steam to air ratio, followed by configurations 2 and 3. Configuration 4 produces less steam than configuration 3 but slightly more power.

DESIGN OPTION 2 RESULTS

The effect of fuel type on the performance of design option 2 is shown in table V. Design option 2 turbomachinery was sized for configuration 3 using the parameters listed in the second column. The basic comparisons shown are again for a steam to air ratio of zero. Configuration 1 was eliminated from this option because it was unable to operate below the compression surge point. The net power output for the remaining configurations were all approximately 20 MW_e. The gas turbine efficiency for the design configuration (intermediate heating value fuel) is approximately the same as obtained using the light distillate. But, again, once the auxiliary losses and gasifier efficiencies are included, the cycle and system efficiencies are higher for the configuration using the light distillate fuel. The cycle and system efficiencies of the low heating value fuel configuration were the lowest of the three cases.

The detailed performance results for option 2 with steam injection are presented in table VI, and in figures 10 to 12. Figure 10 illustrates the effect of steam injection on the net power output for the three configurations. There is not a significant difference in net power output between the configurations at any steam to air ratio. The trend for all cases is again a linear increase in power for increasing injection rates. There is, again, approximately an 8-MW_e increase in power for each 0.1 increase in the steam to air ratio. The values of the steam to air ratio at which compressor surge is encountered and at which the limit of the steam raising capability of the turbine exhaust is encountered are indicated. For configuration 2 the surge limit is encountered before the steam raising limit. For configuration 3 the surge limit and steam raising capability limit both occur at a steam to air ratio of approximately 0.20. For configuration 4 the steam raising limit of the turbine exhaust occurs well before reaching compressor surge. Additional steam is not available for this configuration as it is in the coal-derived fuel gas cases.

Figures 11 and 12 present the cycle and system efficiencies over the applicable range of steam to air ratios. The trend is again increasing cycle and system efficiencies for increased steam to air ratios. The efficiencies are again ranked in the same order as the fuel heating value of each configuation. The larger difference in system efficiency (fig. 11) results because the calculation for light distillate fuel does not have fuel conversion losses.

Figure 13 is a composite of figures 10 and 12 and illustrates the relation between system efficiency, power output, and injection ratio. The trend is again increasing power and efficiency with increased injection ratio, with the magnitude of increase dependent on the configuration. As mentioned in the design option 1 results, the percent increase in efficiency is less than the percent increase in power output because of the additional fuel required for steam injection.

Figure 14 illustrates the relation between the steam available for process use and that used for steam injection. Again, there are two sets of results: one for only the steam raised by reducing the turbine exhaust to the stack limit; the other for the total steam producing capability that includes steam raised in the gasifiers. The total amount of available process steam decreases linearly for increased injection ratios until the compressor surge limit is reached or in the case of configuration 4 (the light distillate case) all the steam raised is used for injection.

Figure 15 illustrates the relation between power, total process steam production, and the amount of steam injection. The trends are similar to those shown for design option 1 (fig. 9) except here configuration 3 is also limited by the compressor surge margin.

CONCLUDING REMARKS

Both the net power output and efficiency of gas turbine systems can be increased by injecting steam into the combustor of the gas turbine. As an increasing amount of steam is injected into the combustor of a fixed geometry gas turbine, the mass flow through the turbine increases relative to the mass flow through the compressor. This increased relative flow increases the compressor back pressure moving the compressor pressure ratio toward the surge line. The maximum increase in performance depends on the surge margin available to the particular compressor.

The turbomachinery characteristics assumed for this analysis had a sufficient surge margin to accommodate a significant amount of steam injection (up to 50 percent of the compressor flow rate). This steam injection did provide substantial increases in both power and efficiency. The percent increase in power is about twice the increase in efficiency for a unit increase in steam injection.

Although the turbomachinery in this analysis was capable of accepting a significant amount of steam, the actual amount of steam available for injection depends on the steam generating capability of the entire power system. Steam can be raised by recovering heat from the turbine exhaust products. This analysis considered gas-turbine systems fueled by either an atmospheric or a pressurized low heating value fuel gasifier, a pressurized intermediate heating value fuel gasifier, or by a conventional light distillate. Steam produced in the gasifiers is also available for injection into the gas-turbine combustor. In all but two of the power systems considered in this analysis, the maximum system steam raising capability (through exhaust heat recovery and gasifier steam) was between 18 and 26 percent of the compressor air flow. This amount of steam injection resulted in power increases of 68 to 100 percent and efficiency increases between 32 and 52 percent over the respective cases without steam injection. The system fueled by a pressurized low heating value fuel gasifier had a system steam raising capability of 37 percent of compressor flow, which results in power and efficiency increases of 145 and 62 percent, respectively, over the noninjected case. The percent increase in performance with steam injection is relatively insensitive to the type of fuel. Steam injection could apply equally well to the various types of integrated-gasifier - gas-turbine systems. However, the use of certain fuels or integration schemes offer a greater potential performance increase.

In an industrial cogeneration application, part of the steam generated by gasifier and the gas-turbine exhaust heat recovery can be used for steam injection, and part can be used to meet industrial process needs. By varying the amount of steam used for injection, the ratio of power to process steam produced by the system can be easily varied. As stated previously, the power output can be varied by up to 145 percent for the cases studied here. When the maximum steam injection rate is used, the steam available for process is zero in all but three cases studied here. (In those three cases the compressor surge line limit was reached before all the steam available was used for injection). When the steam-injection rate is reduced to zero, the steam available for process use is 1769 to 3266 kilograms per hour (3900 to 7200 pounds per hour) per megawatt of power produced for the cases studied.

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TABLE I. - FUEL COMPOSITION FOR EACH CONFIGURATION

Species	Power system configuration			
	1	2	3	4
	Mass fraction			
CO	0.2906	0.2906	0.6033	0
CO_2	.0913	.0913	.2870	
H_2	.0116	.0116	.0281	
CH_4	.0150	.0150	.0685	
N_2	.5851	.5851	.0051	
H_2S	.0005	.0005	.0012	
H_2O	.0059	.0059	.0069	
$CH_{1.942}$	0	0	0	1.0
Higher heating value MJ/Kg (Btu/lb)	5.42 (2332)	5.42 (2332)	13.91 (5982)	43.26 (18 600)

TABLE II. - GAS TURBINE DESIGN OPERATING POINT PARAMETERS

Compressor pressure ratio	11.0
Compressor efficiency (adiabatic)	0.846
Turbine inlet temperature, °C (°F)	1093 (2000)
Turbine pressure ratio	9.735
Turbine efficiency (adiabatic)	0.91
Loss pressure ratio[a]:	0.885
$(\Delta P/P)_{duct}$ 1-2	0.0135
$(\Delta P/P)_{comb.}$ 2-3	0.030
$(\Delta P/P)_{duct}$ 3-4	0.0135
$(\Delta P/P)_{duct}$ 5-6	0.0135
$(\Delta P/P)_{HRSG}$ 6-7	0.050
Mechanical efficiency	0.98
Generator efficiency	0.96
Compressor inlet air flow, $\frac{Kg}{sec}$ $\left(\frac{lbs}{sec}\right)$	69.04 (152.21)
Steam to air ratio	0

[a]Subscripts refer to duct locations shown in fig. 1.

TABLE III. - EFFECT OF FUEL TYPE ON FIXED TURBOMACHINERY SIZE AT DESIGN SPEED - DESIGN OPTION 1 -

	Power system configuration			
	1	2	3	4
Compressir air flow, m_a, Kg/sec (lb/sec)	69.04 (152.21)	69.04 (152.21)	69.04 (152.21)	69.04 (152.21)
Compressor pressure ratio, P_{R_c}	11.0	9.243	9.313	8.987
Compressor efficiency, η_{C_a}	0.846	0.832	0.833	0.827
Turbine pressure ratio, P_{R_t}	9.735	8.253	8.294	8.030
Turbine efficiency, η_{t_a}	0.91	0.91	0.91	0.91
Turbine inlet temperature, T_{IT}, °C (°F)	1093 (2000)	1093 (2000)	1093 (2000)	1093 (2000)
Generator output, W_G, MW_e	30	20.68	21.55	19.45
Net power,[a] W_{net}, MW_e	22.6	19.92	19.14	19.45
Fuel flow, $\dot{m}_f$, Kg/sec (lb/sec)	17.76 (39.15)	15.23 (33.58)	5.27 (11.62)	1.53 (3.37)
Efficiency[b], η, percent:				
Gas turbine	31.2	25.0	29.4	29.4
Cycle	23.5	24.1	26.1	29.4
System	17.8	18.3	21.1	29.4

[a]Generator output minus the power required for motor driven air, oxygen, or fuel gas compressors.

[b]$\eta_{Gas\ Turbine} = W_G/Q_{in\ G.T.}$; $\eta_{cycle} = W_{net}/Q_{in\ G.T.}$; $\eta_{system} = W_{net}/Q_{coal}$, where $Q_{in\ G.T.}$ is the fuel energy input to combustor based on HHV of fuel gas, where Q_{coal} is the coal energy input to gasifier based on HHV of coal, and where, for light distillated, $\eta_{GT} = \eta_{cycle} = \eta_{system}$.

RESULTS - DESIGN OPTION I

Turbine flow to compressor air flow ratio, $\dot{m}_T/\dot{m}_a$	Generator output, MW_e	Auxiliary power requirements, MW_e	Net power output, MW_e	Energy input to combustor, $Q_{in\ G.T.}$, MW_t	Cycle efficiency, η, percent	System efficiency, η, percent	Gasifier air to fuel gas flow ratio, $\dot{m}'_a/\dot{m}_f$	Gasifier oxygen to fuel flow ratio, $\dot{m}_{O_2}/\dot{m}_f$
1.257	30.0	7.4	22.6	96.30	23.5	17.8	0.7628	---
1.371	37.7	8.6	29.1	107.64	27.0	20.5	↓	---
1.482	45.1	10.0	35.1	118.24	29.7	22.6	↓	---
1.537	48.8	10.7	38.1	123.41	30.9	23.5	↓	---
1.052	20.68	0.76	19.92	82.61	24.1	18.3	0.7628	---
1.097	24.04	.77	23.27	87.74	26.5	20.2	↓	---
1.140	27.32	.80	26.52	92.73	28.6	21.7	↓	---
1.182	30.52	.81	29.71	97.57	30.4	23.1	↓	---
1.223	33.64	.82	32.82	102.29	32.1	24.4	↓	---
1.301	39.54	.88	[illegible]	111.33	34.7	26.4	↓	---
1.369	44.83	.93	43.90	119.09	----	28.0	↓	---
1.538	56.89	1.06	55.83	137.55	----	---	↓	---
1.077	21.55	2.41	19.14	73.32	26.1	21.1	---	0.2909
1.132	25.73	2.59	23.14	78.75	29.4	23.7	---	↓
1.188	29.93	2.78	27.15	84.17	32.3	26.0	---	↓
1.243	34.10	2.96	31.14	89.47	34.8	28.1	---	↓
1.299	38.27	3.15	35.12	94.77	37.1	29.9	---	↓
1.409	46.50	3.54	42.96	105.12	----	----	---	↓
1.520	54.53	3.92	50.61	115.09	----	----	---	↓
1.022	19.45	----	19.45	66.16	29.4	29.4	---	---
1.074	23.49	----	23.49	71.08	33.0	33.0	---	---
1.125	27.54	----	27.54	75.98	36.2	36.2	---	---
1.229	35.67	----	35.67	85.56	41.7	41.7	---	---
1.332	43.73	----	43.73	94.86	----	----	---	---
1.435	51.69	----	51.69	103.78	----	----	---	---
1.538	59.40	----	59.40	112.81	----	----	---	---

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TABLE IV. - DETAILED PERFORMANCE

Power system configuration	Compressor pressure ratio, P_{R_c}	Compressor efficiency, η_{c_a}	Compressor air flow, $\dot{m}_a$, Kg/sec (lb/sec)	Combustor air flow to compressor air flow ratio, $\dot{m}_{a_1}/\dot{m}_a$	Gasifier air flow to compressor air flow ratio, $\dot{m}_{a_2}/\dot{m}_{a_1}$	Fuel flow to combustor air flow ratio, $\dot{m}_f/\dot{m}_{a_1}$	Steam to compressor air flow ratio, $\dot{m}_s/\dot{m}_a$
1 - Gas turbine fueled from air-blown atmospheric gasifier	11.0	0.846	69.04 (152.21)	1.0	---	0.257	0
	11.96	.838			---	.287	.0835
	12.91	.818			---	.316	.1657
	13.39	.804			---	.330	.2075
2 - Gas turbine fueled from air-blown pressurized gasifier	9.24	0.832	69.04 (152.21)	0.832	0.168	0.265	0
	9.62	.838		.821	.179	.285	.041 [a](0.05)
	9.98	.842		.811	.189	.305	.081 (0.10)
	10.34	.845		.801	.199	.325	.120 (0.150)
	10.69	.846		.792	.208	.345	.158 (0.20)
	11.37	.843		.773	.227	.385	.230 (0.30)
	12.02	.836		.751	.243	.424	.300 (0.40)
	13.39	.804		.720	.280	.510	.450 (0.626)
3 - Gas turbine fueled from oxygen-blown pressurized gasifier	9.313	0.833	69.04 (152.21)	1.0	---	0.076	0
	9.777	.840			---	.082	.05
	10.242	.844			---	.088	.10
	10.707	.846			---	.093	.15
	11.173	.845			---	.099	.20
	12.106	.835			---	.109	.30
	13.04	.815			---	.120	.40
4 - Gas turbine fueled with a light distillate	8.987	0.827	69.04 (152.21)	1.0	---	0.022	0
	9.424	.835			---	.024	.05
	9.861	.841			---	.025	.10
	10.738	.846			---	.029	.20
	11.618	.841			---	.032	.30
	12.50	.828			---	.035	.40
	13.39	.804			---	.038	.50

[a]Values in parentheses are the steam to combustor air flow ratios.

TABLE V. - EFFECT OF FUEL TYPE ON FIXED TURBOMACHINERY SIZE AT DESIGN SPEED - DESIGN OPTION 2

	Power system configuration		
	2	3	4
Compressor air flow, $\dot{m}_a$, kg/sec (lb/sec)	69.04 (152.21)	69.04 (152.21)	69.04 (152.21)
Compressor pressure ratio, P_{R_c}	12.42	11.0	10.47
Compressor efficiency, η_{C_a}	0.829	0.846	0.845
Turbine pressure ratio, P_{R_t}	11.18	9.735	9.30
Turbine efficiency, η_{t_a}	0.91	0.91	0.91
Turbine inlet temperature, T_{IT}, °C (°F)	1093 (2000)	1093 (2000)	1093 (2000)
Generator output, W_G, MW_e	21	22.2	20.1
Net power,[a] W_{net}, MW_e	20.4	19.9	20.1
Fuel flow, $\dot{m}_f$, kg/sec (lb/sec)	14.40 (31.75)	5.12 (11.28)	1.49 (3.28)
Efficiency[b], η, percent:			
Gas turbine	26.9	31.2	31.3
Cycle	26.1	27.9	31.3
System	19.9	22.5	31.3

[a]Generator output minus the power required for motor driven air, oxygen, or fuel gas compressors.

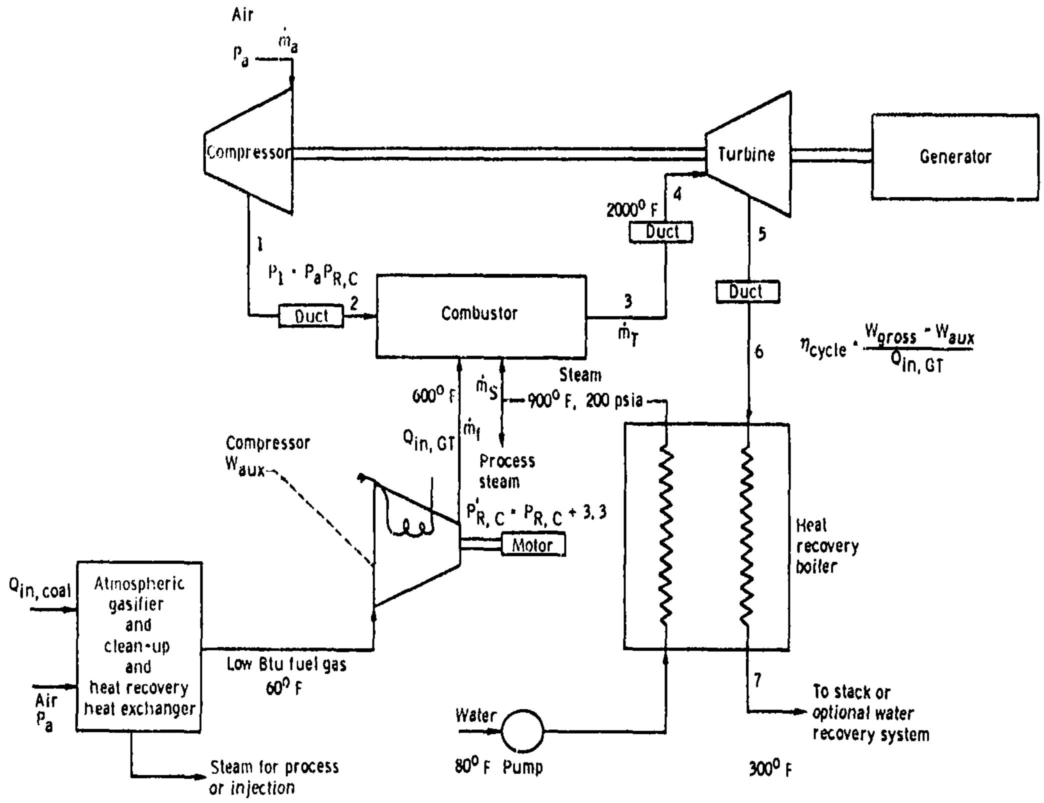
[b]$\eta_{Gas\ turbine} = W_G/Q_{in\ G.T.}$; $\eta_{cycle} = W_{net}/Q_{in\ G.T.}$; $\eta_{system} = W_{net}/Q_{coal}$, where $Q_{in\ G.T.}$ is the fuel energy input to combustor based on HHV of fuel gas, where Q_{coal} is the coal energy input to gasifier based on HHV of coal, and where, for light distillated, $\eta_{GT} = \eta_{cycle} = \eta_{system}$.

RESULTS - DESIGN OPTION 2

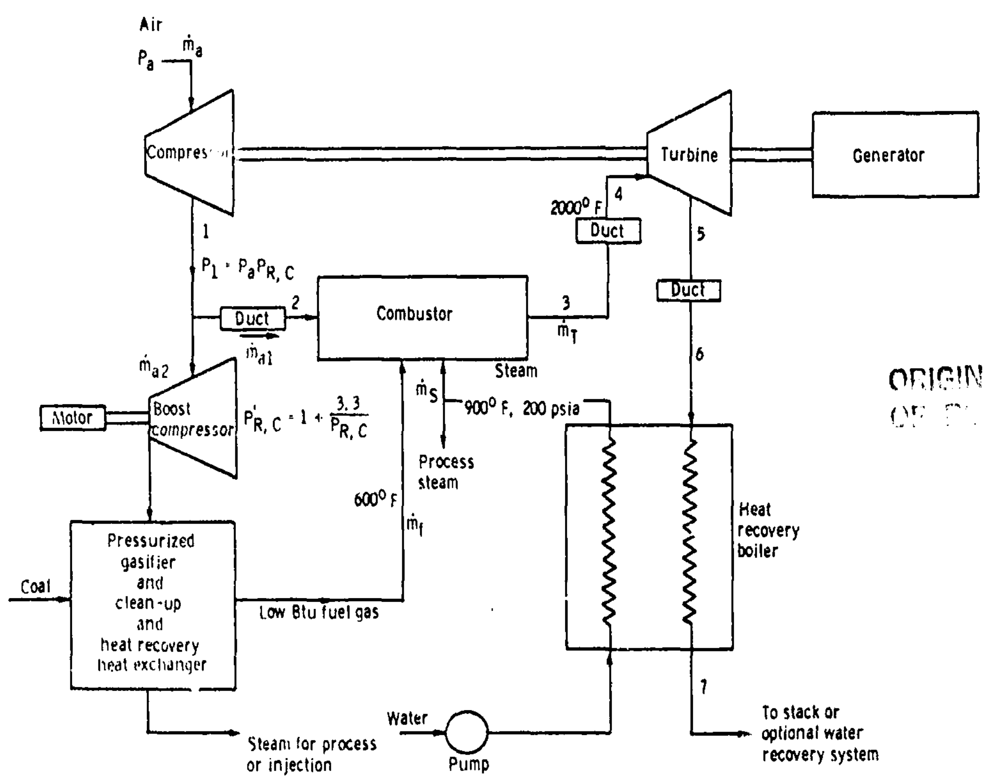
Turbine flow to compressor air flow ratio, $\dot{m}_t/\dot{m}_a$	Generator output, MW_e	Auxiliary power requirements, MW_e	Net power output, MW_e	Energy input to combustor, $Q_{in\ G.T.}$, MW_t	Cycle efficiency, η percent	System efficiency, η percent	Gasifier air to fuel gas flow ratio, $\dot{m}_a/\dot{m}_f$	Gasifier oxygen to fuel flow ratio, $\dot{m}_{O_2}/\dot{m}_f$
1.059	21.0	0.60	20.4	78.18	26.1	19.9	0.7628	---
1.094	24.1	.63	23.47	82.67	28.4	21.6	↓	---
1.132	26.7	.66	26.04	86.54	30.1	22.9		---
1.074	22.23	2.37	19.86	71.17	27.9	22.5	---	0.2909
1.127	26.11	2.55	23.56	76.22	30.9	24.9	---	↓
1.185	30.10	2.72	27.38	81.21	33.7	27.2	---	
1.240	34.0	2.91	31.09	86.07	36.1	29.1	---	
1.294	37.8	3.09	34.71	90.73	38.3	30.9	---	
1.022	20.12	----	20.12	64.35	31.3	31.3	---	---
[illegible]	24.10	----	24.10	69.06	34.9	34.9	---	---
1.125	28.04	----	28.04	73.77	38.0	38.0	---	---
1.176	[illegible]	----	31.93	78.28	40.8	40.8	---	---
1.228	35.76	----	35.76	82.79	----	----	---	---
1.279	39.52	----	39.52	87.11	----	----	---	---
1.311	41.87	----	41.87	89.66	----	----	---	---

TABLE VI. - DETAILED PERFORMANCE

Power system configuration	Compressor pressure ratio, P_{R_c}	Compressor efficiency, η_{C_a}	Compressor air flow, $\dot{m}_a$, kg/sec (lb/sec)	Combustor air flow to compressor air flow ratio, m_{a_1}/m_a	Gasifier air flow to compressor air flow ratio, m_{a_2}/m_{a_1}	Fuel flow to combustor air flow ratio, m_f/m_{a_1}	Steam to compressor air flow ratio, $\dot{m}_s/\dot{m}_a$
2 - Gas turbine fueled from an air-blown pressurized gasifier	12.42	0.829	69.04 (152.21)	0.841	0.159	0.248	0
	12.94	.817		.831	.168	.266	.042
	13.39	.804		.824	.176	.281	.077
3 - Gas turbine fueled from oxygen-blown pressurized gasifier	11.0	0.846	69.04 (152.21)	1.0	---	0.074	0
	11.55	.842			---	.079	.05
	12.11	.835			---	.085	.10
	12.66	.824			---	.090	.15
	13.21	.810			---	.094	.20
4 - Gas turbine fueled with a light distillate	10.47	0.845	69.04 (152.21)	1.0	---	0.022	0
	10.99	.845			---	.023	.05
	11.50	.842			---	.025	.10
	12.02	.836			---	.026	.15
	12.54	.827			---	.028	.20
	13.06	.814			---	.029	.25
	13.39	.804			---	.030	.28

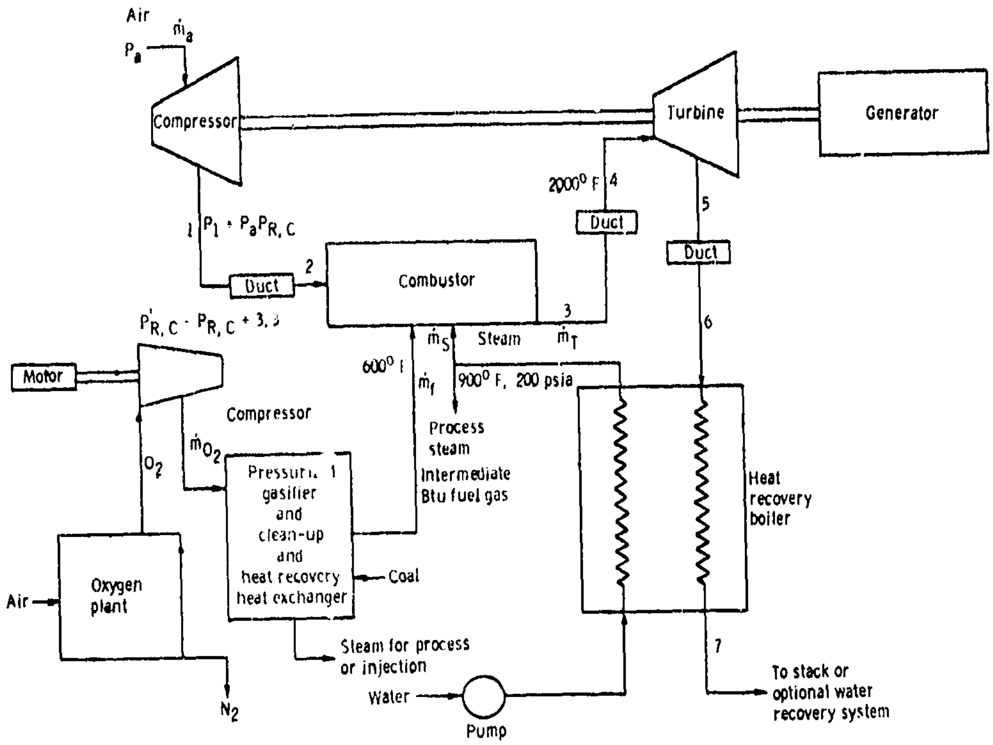


(a) Configuration 1: gas turbine with steam injection fueled from an air-blown atmospheric gasifier.

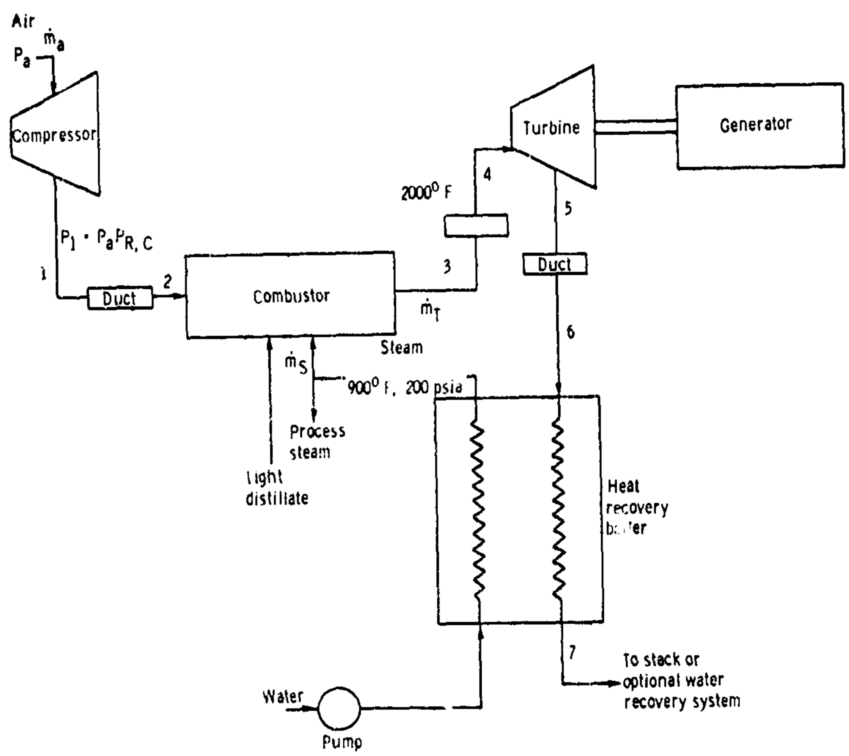


(b) Configuration 2: gas turbine with steam injection fueled from an air-blown pressurized gasifier.

Figure 1. - System schematics.

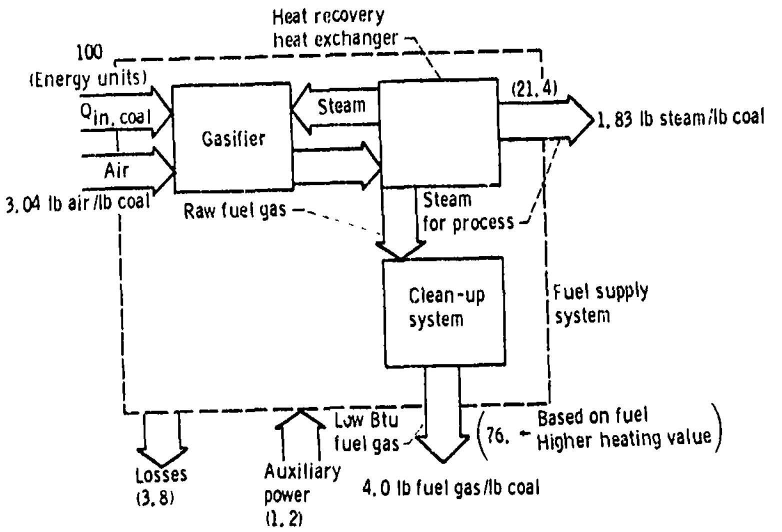


(c) Configuration 3: gas turbine with steam injection fueled from an oxygen-blown pressurized gasifier.

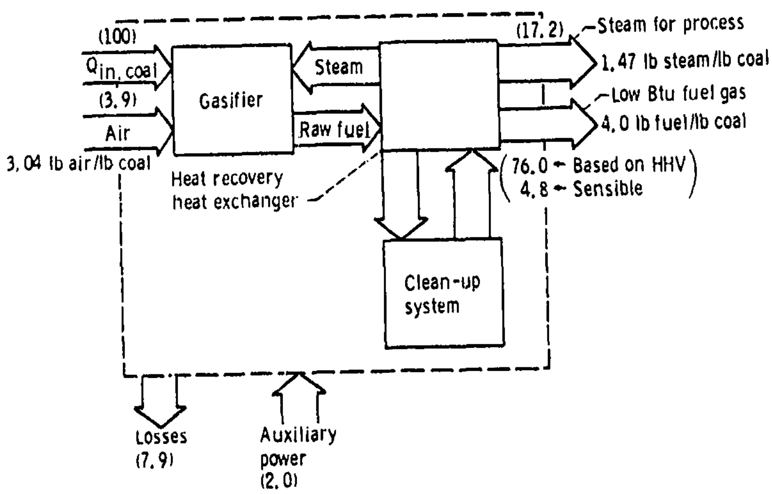


(d) Configuration 4: gas turbine with steam injection fueled with a water distillate.

Figure 1. - Concluded.

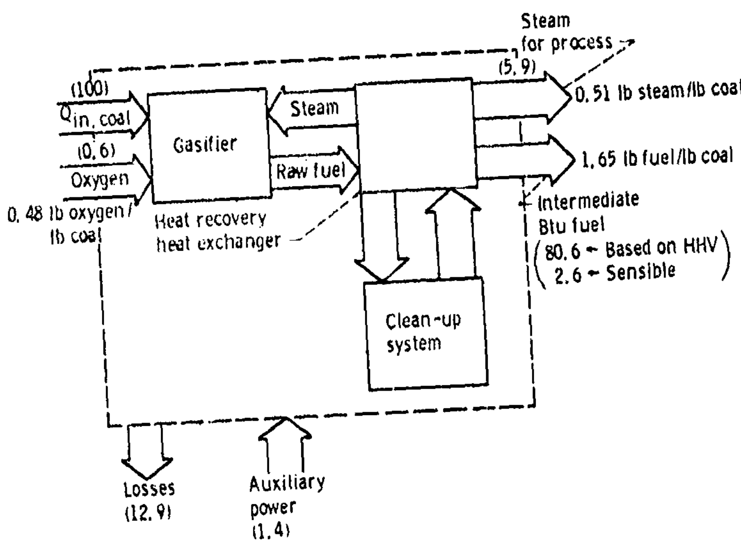


(a) Configuration 1: atmospheric gasifier, low Btu fuel.



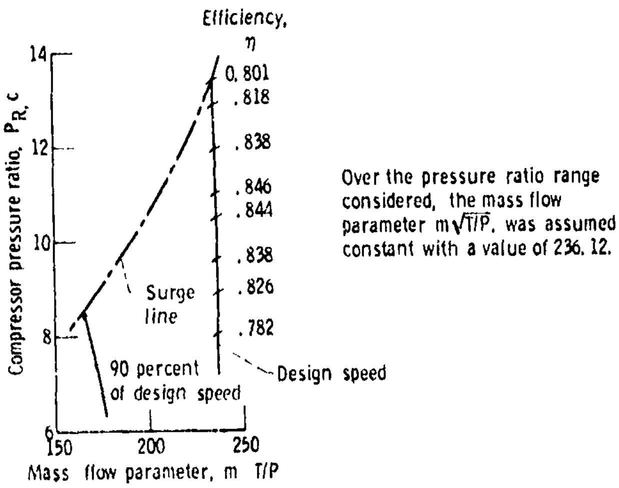
(b) Configuration 2: pressurized gasifier, low Btu fuel.

Figure 2. - Energy flow paths for fuel supply systems of configurations 1, 2, and 3.

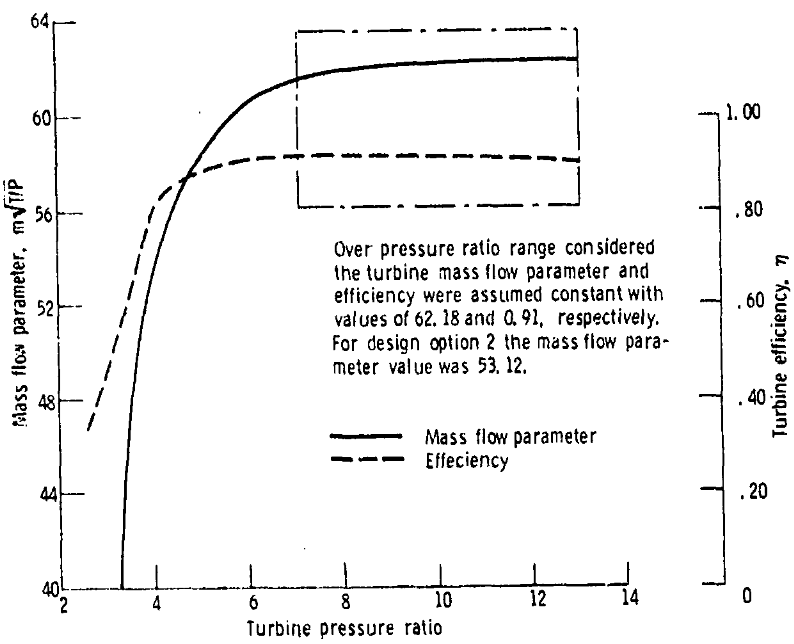


(c) Configuration 3: pressurized gasifier, intermediate Btu fuel.

Figure 2. - Concluded.



(a) Compressor



(b) Turbine

Figure 3. - Compressor and turbine characteristics for design options 1 and 2.

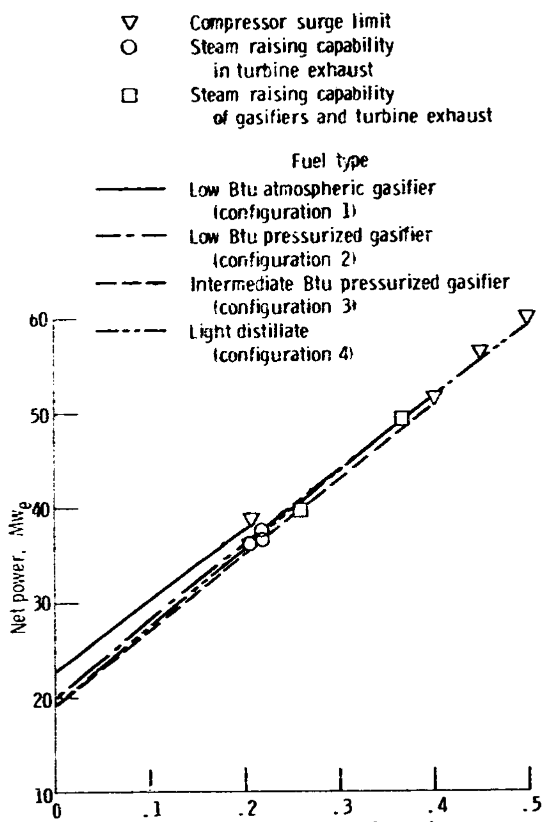


Figure 4. - Effect of fuel type and steam injection on net power output for fixed turbomachinery size of design option 1.

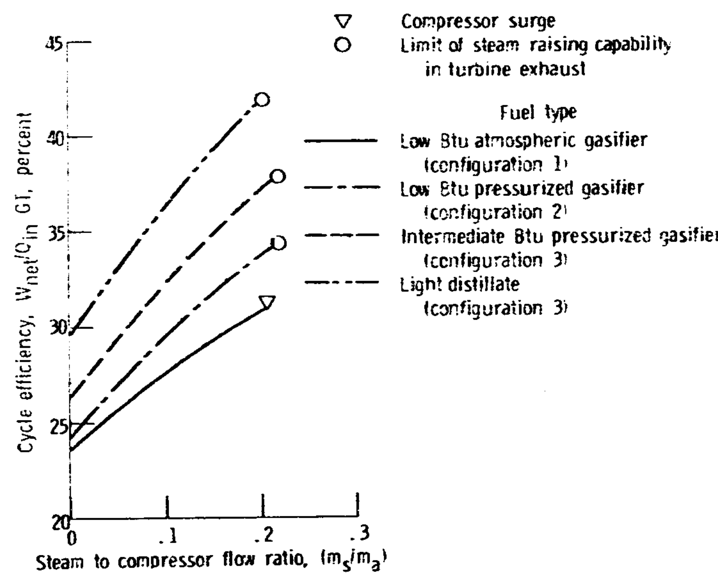


Figure 5. - Effect of fuel type and steam injection of cycle efficiency for the fixed turbomachinery size of design option 1.

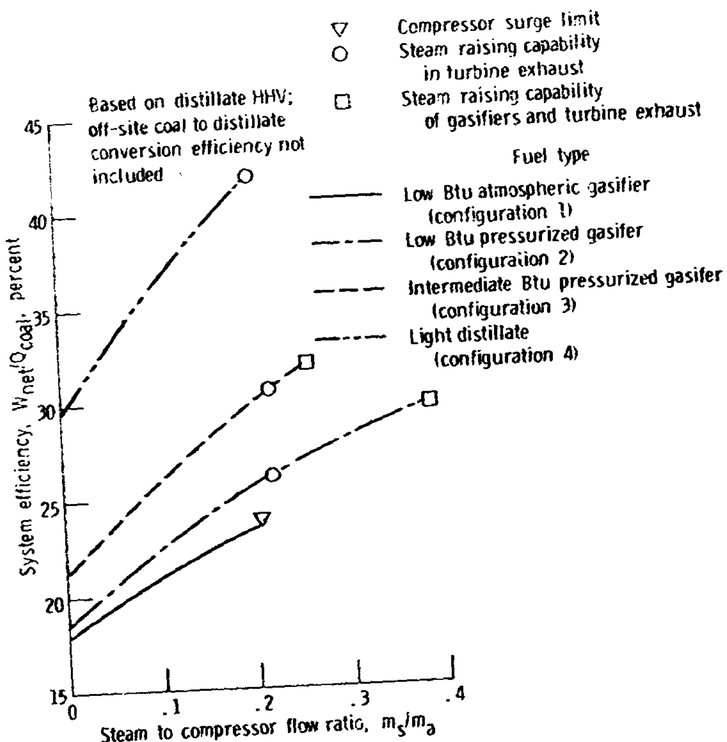


Figure 6. - Effect of fuel type and steam injection on system efficiency for the fixed turbomachinery size of design option 1.

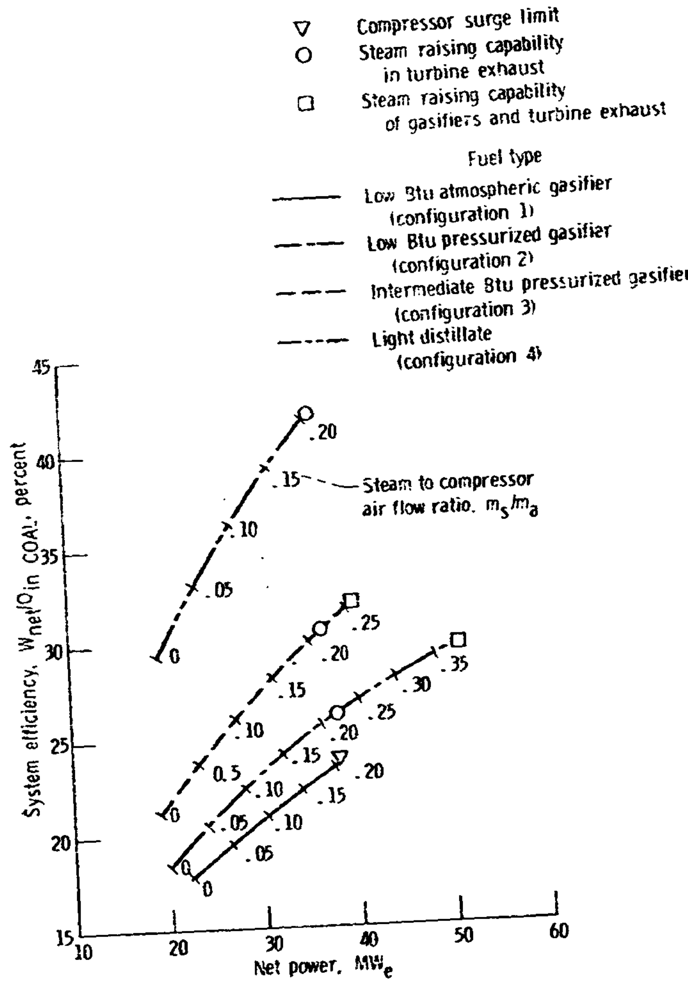


Figure 7. - Performance of steam injection gas turbine power systems for the fixed turbomachinery size of design option 1 for various fuels.

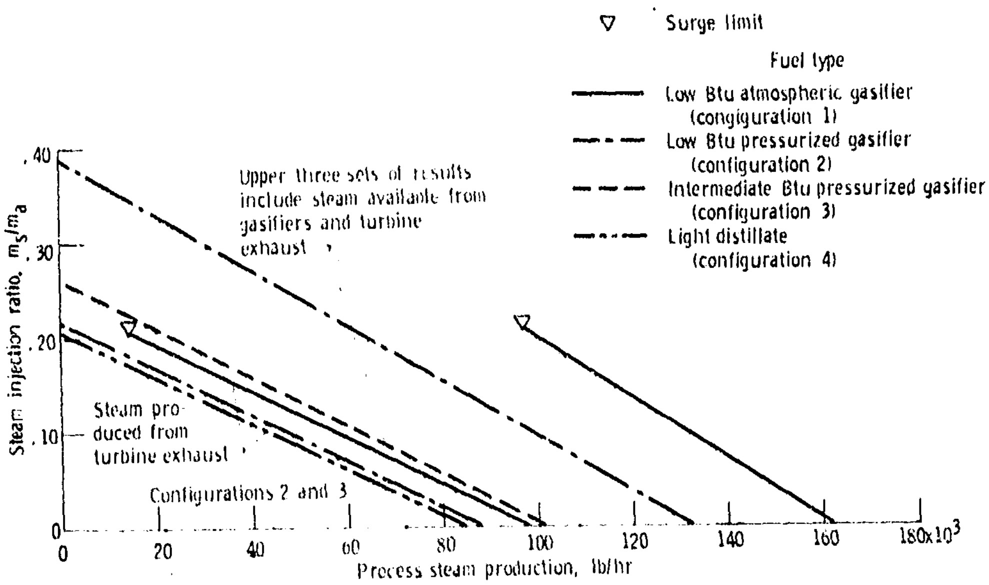


Figure 8. - Relation between steam used to increase power and that available for process use. Steam temperature and pressure, 900° F and 200 psi.

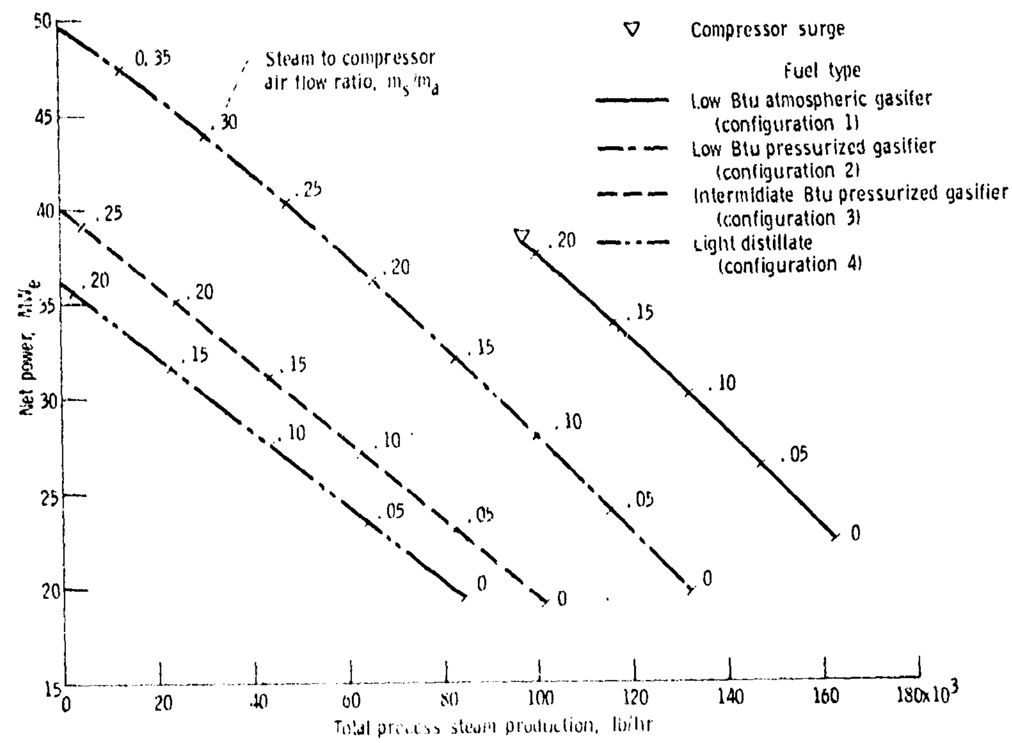


Figure 9. - Cogeneration system performance for the fixed turbomachinery size of design option 1 for the various fuels. Steam temperature and pressure, 900° and 200 psi.

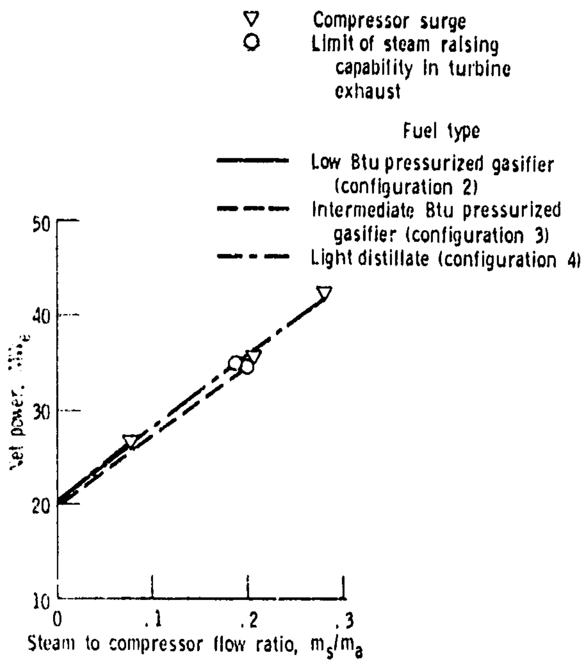


Figure 10. - Effect of fuel type and steam injection on net power output for the fixed turbomachinery size of design option 2.

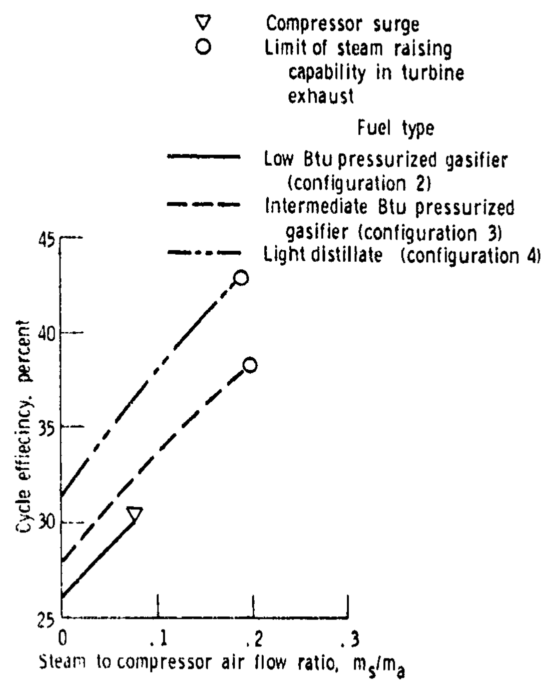


Figure 11. - Effect of fuel type and steam injection on cycle efficiency fixed turbomachinery size of design option 2.

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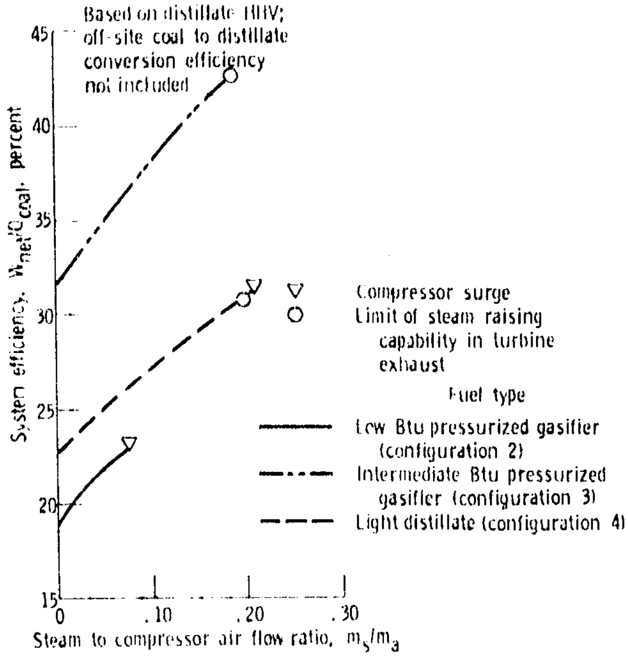

Figure 12. - Effect of fuel type and steam injection on system efficiency for the fixed turbomachinery size of design option 2.

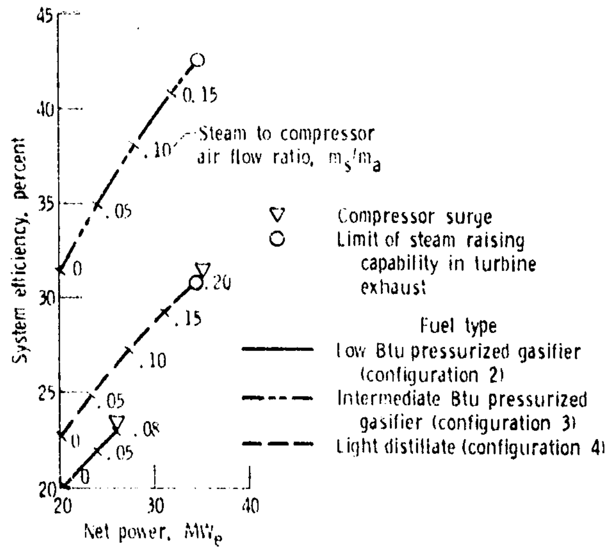

Figure 13. - Performance of gas turbine power cycles for the fixed turbomachinery size of design option 2 for various fuels and steam-injection ratios.

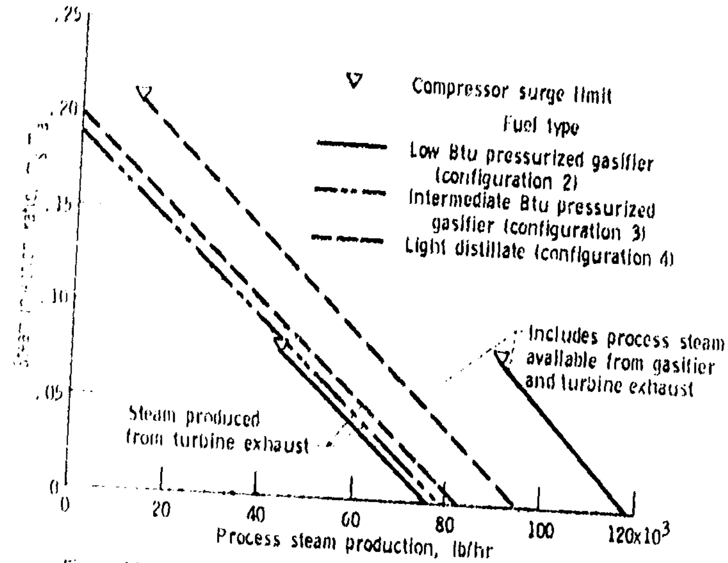

Figure 14. - Relation between steam used to increase power and that available for process use. Design option 2; steam temperature and pressure, 900° F, and 200 psi.

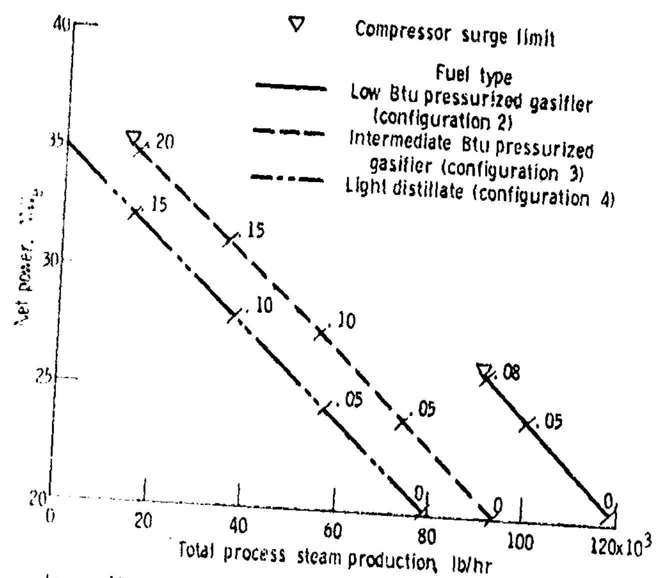

Figure 15. - Cogeneration system performance for the fixed turbomachinery size of design option 2 for the various fuels. Process stream temperature and pressure, 900° F and 200 psi. Total process steam includes that availabe from gasifier and turbine exhaust.

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15. Supplementary Notes

16. Abstract

The results of an analysis to estimate the off-design performance of a specific gas turbine powerplant augmented by steam injection are presented. Results are compared using coal-derived low and intermediate heating valve fuel gases and a conventional distillate. The results indicate that steam injection could provide substantial increases in both power and efficiency within the available compressor surge margin. The results also indicate that these performance gains are relatively insensitive as to the type of fuel.

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