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THE INTEGRATED APPROACH TO A GAS TURBINE TOPPING CYCLE COGENERATION SYSTEM

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ABSTRACT

Under Gas Research Institute (GRI) sponsorship, a new gas turbine cogeneration system was developed by Mechanical Technology Inc. (MTI) for installation at a General Motors plant in early 1985. Specific emphasis was placed on system integration. A single, prime-reliable drive train and a single control center replace a wide assortment of nonintegrated, free-standing power drives and control centers. On-line availability, installation costs, and overall user acceptance are improved.

The cogeneration set produces 3 MW_e and 8,860 kg/hr (19,500 lb/hr) of 1825 kPa (250 psig) saturated steam using an Allison 501-KH gas turbine and a natural circulation waste heat boiler. The system is designed for multifuel operation using either natural gas or distillate oil. A steam injection feature is employed to increase output to 4 MW_e when process steam demand diminishes. The system is prepackaged, skid mounted and delivered in four modules; one each for the machinery, duct burner, waste heat boiler, and controls.

INTRODUCTION

Although the rules and definitions that govern eligible cogenerators have been in existence since PURPA of 1978, the marketplace has been conspicuously slow in accepting this seemingly better approach to energy utilization. In the entire state of California, where cogeneration has been actively promoted, there are approximately only thirty-five systems that are operational at this time. Although some point to the poor ratio of electric/fuel prices as the culprit, there is more reason to believe that the major factor causing weak market penetration is poor system integration and the user's perception of the complexity and "unfriendliness" that follows.

An advanced gas turbine topping cycle cogeneration system has been developed by Mechanical Technology Inc., with support from the Gas Research Institute (GRI). GRI is a gas utility-financed organization that was established in 1977 to enhance and promote the use of natural gas through externally funded research projects. The major goal of this project was to develop an integrated system with enhanced performance through the implementation of several engineering development tasks, namely:

- Control integration
- Mechanical equipment integration
- Rotary screw gas compressor application
- Steam injection application.

The belief is that well-integrated, efficient, easy-to-operate systems are "user friendly" and, as such, will gain better marketplace acceptance.

The prototype cogeneration system will be installed at a General Motors plant in early 1985. It consists of an Allison 501-KH steam-injected gas turbine and a supplementary-fired waste heat boiler. Nominally, the system produces 3 MW_e and 8,860 kg/hr (19,500 lb/hr) of 1825 kPa (250 psig) saturated steam. Supplemental firing increases the steam production to 15,900 kg/hr (35,000 lb/hr). When thermal demand falls off, steam is injected into the gas turbine combustor to increase electrical output and to improve the system's heat rate. At 2.27 kg/sec (5 lb/sec) of steam injection, the system's output increases to 4 MW_e and heat rate drops by 20% to 11,300 kJ/kWh (10,700 Btu/kWh).

The design strategy, system and component specifications, and estimated system performance levels are the subject of this paper.

SYSTEM DESIGN APPROACH

The system has been specifically designed for industrial cogeneration service, with an emphasis on fitness for duty, modularity and reliability. This is a major departure from the conventional approach where components, originally specified for other applications, have been specified and arranged in an expedient, suboptimum manner. As a consequence, conventional systems suffer from a redundancy of parts and operations, are more expensive to operate, and suffer in terms of their on-line availability.

The subject system contains components that have been selected or designed for a unique cogeneration function, and are integrated as much as possible on a prepackaged skid. System reliability has also been thoroughly addressed. Cogeneration systems are expected to operate at base load, imposing a greater availability requirement than that for peaking or pump station power systems. Accordingly, the subject system is designed with a single, prime-reliable drive system. All of the major components that transmit or develop power operate within the same drive train. The load gearbox contains six shaft penetrations; one each for the gas turbine, electric generator, fuel gas compressor, lube oil pump, boiler feed pump and turbine starter. See Figure 1. The electrical and/or mechanical prime movers and support accessories normally used in conjunction with the gas compressor, boiler feed pump, and lube oil pump, along with their attendant unavailabilities, have been eliminated.

Another feature which enhances integration and system reliability is the gas turbine-driven fuel gas compressor. Instead of contending with a separate skid-mounted, diesel-engine-driven reciprocating compressor, the compressor selected for the subject system is a compact rotary screw type driven directly by the gearbox and located on the same skid as the gas turbine. Approximately 13.8 m^2 (150 ft^2) of skid surface is saved by eliminating a redundant cooling system, motor control center, support structure, and piping. The rotary screw compressor is also much smaller and more reliable than its reciprocating counterpart.

The third system feature which addresses the integration issue, and perhaps the most significant feature from an operational consideration, is the overall system control. Controls that normally accompany each major subsystem (turbine generator, waste heat boiler, duct burner and gas compressor) as separate panels and displays are now fully merged into a single cabinet and control center. A centrally located motor control center, process controller, annunciator panel and instrument display console will service the entire cogeneration package. To accommodate a variety of siting requirements, the control skid is prepackaged and may be located adjacent to the machinery skid or at a remote location designated by the user.

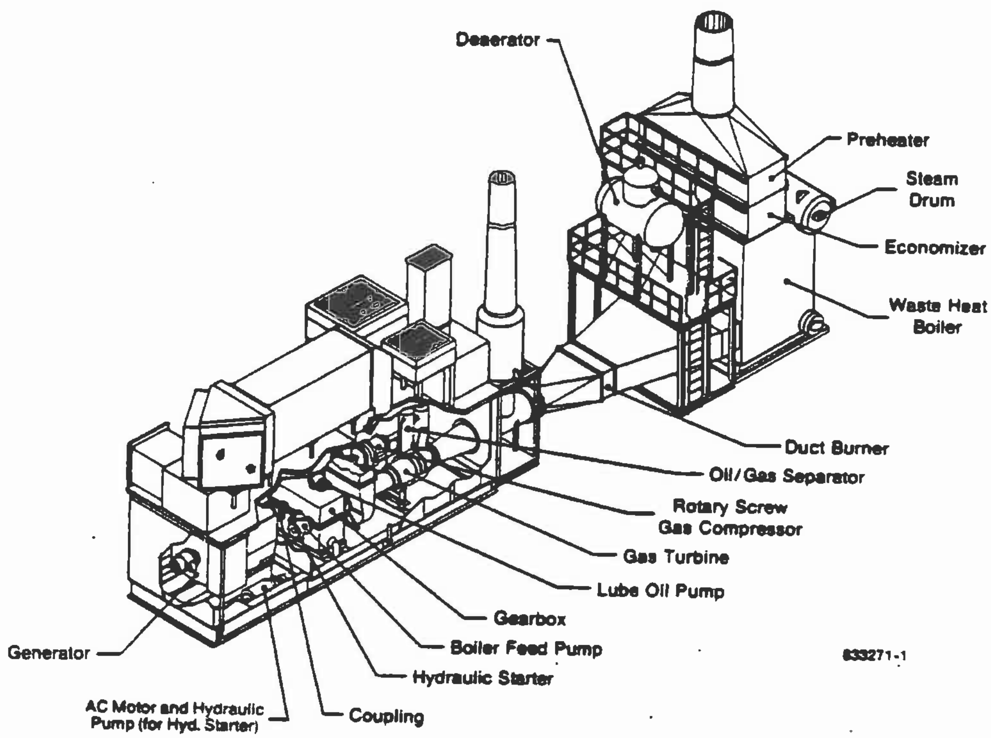

FIG. 1 COGENERATION SYSTEM - SKID ARRANGEMENT

SYSTEM SPECIFICATION

The cogeneration system is shown schematically in a process flow diagram, Figure 2. The energy and mass balance state points for simple cycle and steam injection operation are presented in Table 1. The addition of 1.82 kg/sec (4 lb/sec) of steam increases the compressor pressure ratio from 9.3 to 10.3 but decreases the turbine inlet temperature (TIT) to 913°C (1675°F). If an additional 4.4×10^6 kJ/hr (4.2×10^6 Btu/hr) of fuel had not accompanied the steam injection, the TIT would have been reduced even further to approximately 871°C (1600°F).

PERFORMANCE WITH VARIABLE STEAM DEMAND

The performance maps shown in Figure 3 illustrate the performance of the system as it varies with steam delivered to process. Two possible modes of operation are presented; one at constant fuel flow (points a, b, and c) and the other at the high power setting (points a, e, and d). At constant fuel flow, the injected steam (W_s) causes the TIT to drop by approximately 12.6°C/kg/sec (50°F/lb/sec). Nevertheless, output and heat rate are both improved; the output increases to 3630 kW and the heat rate decreases to 11,700 kJ/kWh (11,100 Btu/kWh) at full injection (no steam to process).

The operator also has the option of maintaining the TIT as steam is injected. This results in the highest power setting. At a steam delivery of 5640 kg/hr (12,400 lb/hr), shown as point e, the slope of the curve changes as the control limit parameter changes from TIT to generator output. Along this mode, the output and heat rate at full injection reach 4 MW_e and 11,300 kJ/kWh (10,700 Btu/kWh), respectively.

SYSTEM SIZING

Although the 501-KH engine has a torque limit corresponding to 6190 kW (8300 hp) (refer to later section on Engine Performance), its maximum rated capacity for this system is 4850 kW (6500 hp). Derating the system to 4000 kW_e was a result of a conscious decision to optimize user economics instead of output. To the extent that this system deals primarily with the cogeneration (electric and thermal output) application, and secondarily with the power-only mode of operation, the period of operation at full injection was considered too short to justify increasing the power rating and corresponding cost of the system's major components, e.g., generator, gearbox, fuel gas compressor, starter, etc. Furthermore, operation at levels substantially below capacity increases parasitic losses and, likewise, the system's heat rate; this was another factor which favored the 4 MW_e power limit.

GENERAL ARRANGEMENT

The system is housed on four skids: one each for the machinery, waste heat boiler, duct burner, and one for the controls and electric switchgear. The machinery, duct burner and waste heat boiler skids are shown in Figure 4. Together, the skids measure 20.4 m long by 3.25 m wide (67 ft by 10 ft 8 in.).

The integrated drive train featuring the six shaft penetration gearbox and the turbine-driven gas compressor is shown pictorally in Figure 1. A clutch is installed between the gas compressor and gearbox to uncouple the compressor during start-up and at all other times when operating on distillate oil. The clutch requirement is explained in greater detail in the discussion of the system start-up procedure in a later paragraph.

Due to the close proximity of the gas compressor to the other machinery, a fire wall is installed to separate the generator compartment from the gas turbine and auxiliary machinery. Positive pressure is maintained in the generator compartment with forced draft fans to ensure that no migration of natural gas into the generator compartment can occur. Moreover, an induced draft fan, located behind the gas turbine diffuser, cools the turbine case and evacuates any gas leaks overboard.

The duct burner and waste heat boiler are located directly downstream of the gas turbine diffuser. Provisions have been made in the skid design to accommodate a diverter valve, although this is not shown as a bill of material item. Since the system is expected to generate steam at all times, either for process use or for turbine injection, there is little or no need for a bypass stack. Furthermore, the waste heat boiler specification allows for operation during nonsteaming periods. A level of water will be maintained inside the drum during start-up to minimize tube wall temperatures and weld stresses. Still, if the customer so chooses, a bypass stack can be added without difficulty.

STEAM INJECTION

Cogeneration systems normally operate on a thermal load following basis (steam demand). Large swings in steam demand, both diurnal and annual, may have significant impact on the economic viability of the installation. An effective way to minimize these "swings" and to keep the gas turbine base loaded is to inject excess steam into the gas turbine when process steam demand is low. Although this particular application is somewhat different, the art and operation of industrial gas turbines with steam injection is not new. Bultzo at Exxon Baytown reported on the performance improvement of the GE MS 3000 with steam injection in 1969 (1), while Kydd and Day (1972) (2) described a system wherein steam generated in an unfired waste heat boiler was injected into a gas turbine.

The subject system contains this feature; the Allison 501-KH is the steam-injected version of the 501-KB. Other than providing steam manifolds on the outer combustor case and strengthening the first-stage compressor disc and output shafts, no other turbine structural modifications were necessary.

The choice of steam injection conditions was made on the basis of performance, cost and end-user familiarity. Market studies showed that 1825 kPa (250 psig), 208°C (406°F) dry saturated steam provided the best market coverage. While injecting with superheated steam improves performance (heat rate) over that with saturated steam by about 6% during power-only operation, the selection of the latter over the former results in substantial benefits to the end user. As stated previously, the product addresses the cogeneration duty cycle as its primary application with the power-only mode being less important. That places process requirements first, and hence the bias toward saturated steam. Selecting saturated steam over superheated steam also eliminates the cost of a superheater and a desuperheating station, the attendant superheater unavailability, the additional water treatment requirements and, perhaps most significantly, the end user's lack of familiarity (and corresponding negative perception) with the additional complexity imposed by a superheater. Finally, Allison has completed its shop tests (3) on the 501-KH using 1825 kPa saturated steam. With due consideration given to all of the above, the decision was made to inject dry saturated steam at 1825 kPa.

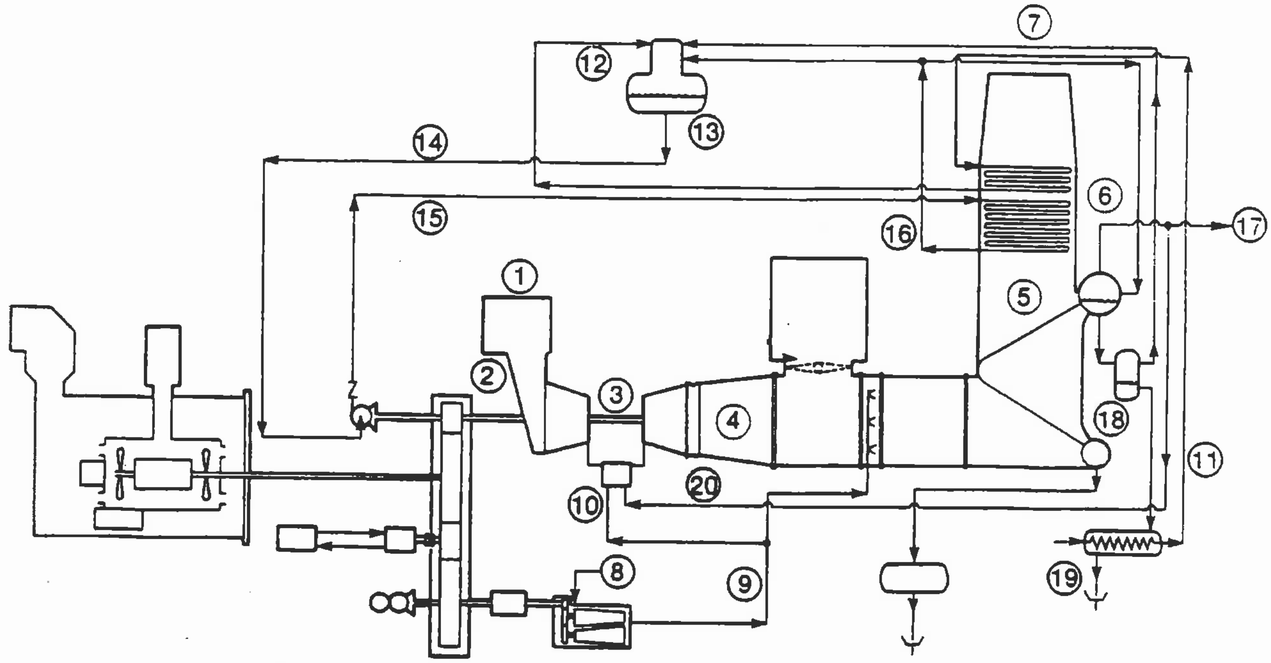

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FIG. 2 COGENERATION SYSTEM ENERGY AND MASS BALANCE

TABLE 1 THERMODYNAMIC STATE POINTS ENERGY AND MASS BALANCE

		Without Steam Injection						With Steam Injection					
	Station	Flow		Pressure		Temperature		Flow		Pressure		Temperature	
		kg/hr	(lb/hr)	kPa	(psia)	°C	(°F)	kg/hr	(lb/hr)	kPa	(psia)	°C	(°F)
Air/Gas	1	55,950	(123,100)	101	(14.7)	16	(60)	55,950	(123,100)	101	(14.7)	16	(60)
Air/Gas	2	55,950	(123,100)	100	(14.6)	16	(60)	55,950	(123,100)	100	(14.6)	16	(60)
Air/Gas	3	56,790	(124,950)	944	(137.0)	982	(1800)	63,420	(139,500)	1033	(149.9)	913	(1675)
Air/Gas	4	↓	↓	103	(14.9)	510	(950)	↓	↓	103	(14.9)	456	(852)
Air/Gas	5			—	—	222	(431)			—	—	226	(439)
Air/Gas	6			—	—	161	(322)			—	—	178	(352)
Air/Gas	7			101	(14.7)	136	(277)			101	(14.7)	152	(305)
Fuel	8	839	(1845)	586	(84.7)	16	(60)	926	(2037)	586	(84.7)	16	(60)
Fuel	9	839	(1845)	2067	(299.7)	54	(130)	926	(2037)	2067	(299.7)	54	(130)
Fuel	10	Q_{GT} = 42.4 × 10^6 kJ/hr		(40.2 × 10^6 Btu/hr)				Q_{GT} = 46.8 × 10^6 kJ/hr		(44.4 × 10^6 Btu/hr)			
Steam	11	9,180	(20,200)	446	(64.7)	66	(150)	8,909	(19,600)	446	(64.7)	66	(150)
Steam	12	↓	↓	446	(64.7)	107	(225)	↓	↓	446	(64.7)	107	(225)
Steam	13			205	(29.7)	121	(250)			205	(29.7)	121	(250)
Steam	14			136	(19.7)	121	(250)			136	(19.7)	121	(250)
Steam	15			2,857	(414.7)	121	(250)			2857	(414.7)	121	(250)
Steam	16			2,857	(414.7)	191	(375)			2857	(414.7)	191	(375)
Steam	17	8,863	(19,500)	1825	(264.7)	208	(406)	2,045	(4,500)	1825	(264.7)	208	(406)
Steam	18	318	(700)	205	(29.7)	121	(250)	318	(700)	205	(29.7)	121	(250)
Steam	19	318	(700)	101	(14.7)	100	(212)	318	(700)	101	(14.7)	100	(212)
Steam	20		No Injection					6,545	(14,400)	1825	(264.7)	208	(406)
		Net Generator Output 3000 kW_e						Net Generator Output 4000 kW_e					
		System Heat Rate (LHV) 14,120 kJ/kWh (13,400 Btu/kWh)						System Heat Rate (LHV) 11,700 kJ/kWh (11,100 Btu/kWh)					

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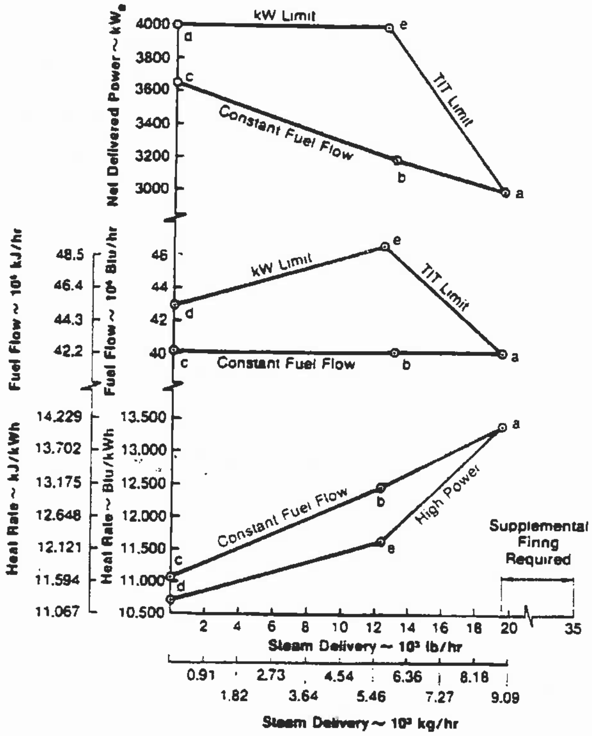

FIG. 3 COGENERATION SYSTEM PERFORMANCE - 16°C (60°F) DAY, GAS FUEL AT 583 KPA (70 PSIG)

ENGINE PERFORMANCE

The performance characteristics of the 501-KH, shown in Figure 5, describe power and efficiency at the output shaft as a function of steam injection flow rate and TIT. The limits of engine operation are denoted by the torque limit line at 6190 kW (8300 hp) and the line of maximum steam flow. At maximum steam flow, the engine is operating at a TIT at which all steam generated in the waste heat boiler (without supplemental firing) is injected into the engine.

At 982°C TIT, 2.3 kg/sec, ISO, the engine's output increases by 82%, from 3394 kW to 6190 kW (4550 hp to 8300 hp). The corresponding improvement in engine efficiency is 32%. Operating in the subject 4 MW_e system, however, the engine's output is limited to 4850 kW (6500 hp). Maximum engine efficiency drops to 36.0%, about 4% lower than at the 6190 kW setting, but still 26% better than the simple cycle (baseline) case.

The shop tests conducted by Allison in 1982 have confirmed these performance improvements. Saturated steam at 1825 kPa (250 psig) was injected into the combustor of a "workhorse" 501-KB engine at two axial locations; one at the combustor head end and one near the tail end. Agreement between analytical predictions and measured values was quite good. At all levels of injection below 2.3 kg/sec, the predicted and measured values were virtually coincidental with each other. At 2.3 kg/sec, the measured power was 1% to 3% lower than predicted. No discernible difference was measured due to injection locations, i.e., upstream injection, downstream injection and a combination of both showed good agreement except at 2.3 kg/sec as noted.

A post-mortem examination of the turbine hardware and engine lube oil system showed no adverse effects due to steam injection.

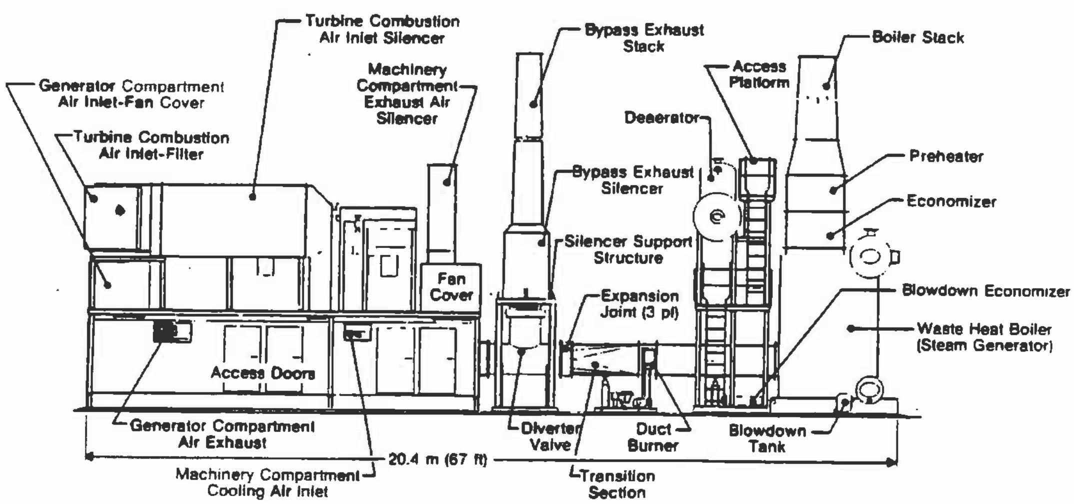

FIG. 4 COGENERATION SYSTEM - ELEVATION VIEW

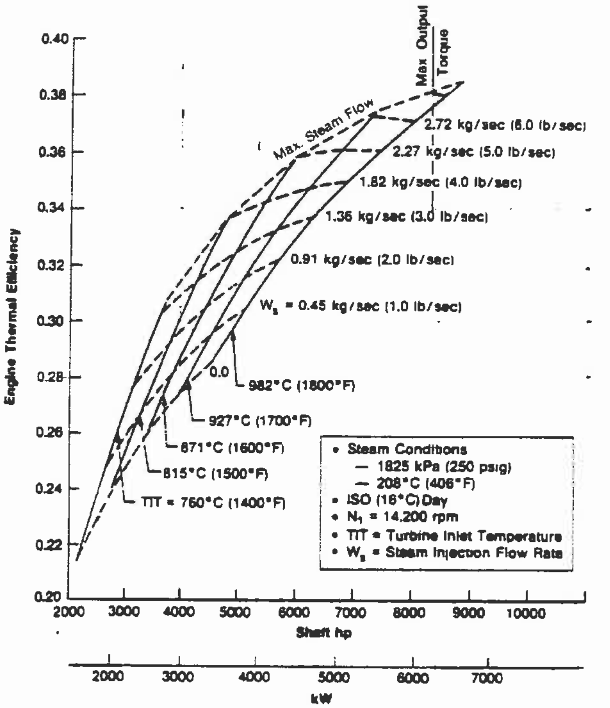

FIG. 5 501-KH ENGINE PERFORMANCE WITH STEAM INJECTION

MAJOR COMPONENTS

Gas Turbine

The Allison 501-KH gas turbine was selected because it is particularly well suited for cogeneration duty with consideration given to size, heat rate, steam generating capacity, reliability/availability, and capital cost. Moreover, with a compressor stall margin in excess of 45%, this engine is ideal for steam injection. Injecting steam at a steam/air mass ratio of 0.12 increases the compressor's pressure ratio approximately 20%. This would impose a severe restriction on engines with stall margins of only 20 to 25%, whose compressors would require extensive redesign for steam injection.

The 501-KH is the steam injected version of the popular 501-KB. It is the commercial version of the T56-501K turboprop engine of which 14,000 have been built and have logged in excess of 100 million hours of operation. It is a single shaft engine utilizing a 14-stage axial flow compressor, 6 combustion chambers within an annular combustor, and a 4-stage turbine with an air-cooled first stage. The compression ratio is approximately 9.3:1 at standard continuous rating. The five main antifriction bearings are pressure lubricated. A dual fuel (gas and distillate oil) version was selected for cogeneration duty.

Two minor hardware changes accompany the transition from the KB to the KH version. Two steam manifolds are welded to the outer case surrounding the combustion liners; one at the combustor head and one downstream in the transition section. Steam entering the case through these manifolds mixes with the compressor discharge air and the steam/air mixture is metered through the same combustor liner holes used for air-only metering.

The other modification consists of strengthening the first stage compressor disc, extension shaft and power takeoff (PTO) shaft to accommodate the additional torque.

In addition to these structural changes, the 501-KH is equipped with an incipient stall detection system to prevent stall due to excessive amounts of steam injection.

Waste Heat Boiler

A natural circulation waste heat boiler was specified for the subject system. It consists of a vaporizer, economizer and liquid preheater. Based on a gas turbine exhaust flow of 15.6 kg/sec (34.2 lb/sec) at 510°C (950°F), the steam generator produces 8,860 kg/hr (19,500 lb/hr) of 1965 kPa (270 psig) saturated steam. The waste heat boiler is also designed to accommodate up to 18.2 kg/sec (40.0 lb/sec) of flow during periods of steam injection into the turbine. A maximum gas side total pressure drop of 8.2 mm Hg was specified. The specified water and steam conditions are presented on Table 2 for simple cycle and steam injection operation.

Feedwater heating is accomplished by a separate deaerating feedwater heater which withdraws saturated liquid from the outlet of the economizer.

Supplemental Firing. A duct burner is employed to increase the steam generation to 15,900 kg/hr (35,000 lb/hr). No refractory material is used. A maximum duct gas temperature of 760°C (1400°F) is specified.

TABLE 2 WASTE HEAT BOILER WATER/STEAM SPECIFICATION

Component	Temperature °C	Temperature (°F)	Pressure kPa	Pressure (psig)	Simple Cycle Flow kg/hr	Simple Cycle Flow lb/hr	Steam Injection Flow kg/hr	Steam Injection Flow lb/hr
Preheater								
Inlet Water	65	(150)	274	(25)	9,182	(20,200)	8,136	(17,900)
Outlet Water	107	(225)	205	(15)				
Maximum Allowable Pressure Drop			69	(10)				
Economizer								
Inlet Water	121	(250)	2168	(300)	9,182	(20,200)	8,136	(17,900)
Outlet Water	206	(403)	203	(280)				
Maximum Allowable Pressure Drop			172	(25)				
Vaporizer								
Steam Discharge	212	(413)	1962	(270)	8,863	(19,500)	7,864	(17,300)
Continuous Blowdown (est)					318	(700)	273	(600)
Steam Quality	99.5%							

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Fuel Gas Compressor

As stated previously, the integration of the fuel gas compressor into the main turbine skid represents one of the system's most significant departures from conventional cogeneration packaging practice. Integration within the system required a compact, high-pressure ratio, low-flow device whose rotational speed was compatible with the other skid machinery. After thoroughly examining several candidate technologies, the selection rapidly converged on a rotary screw compressor (see Figure 6).

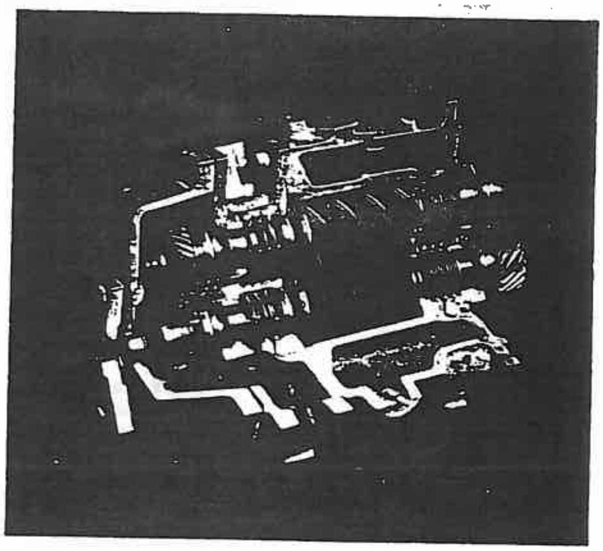

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FIG. 6 ROTARY SCREW FUEL GAS COMPRESSOR

The compressor can deliver 22.2 standard m^3/hr (783 scfm) of natural gas at 2067 kPa (285 psig) starting with a suction pressure of 138 kPa (5 psig). It provides 49.5 x 10^6 kJ/hr (47 x 10^6 Btu/hr) of fuel to the gas turbine while consuming approximately 224 kW (300 hp) of shaft power.

The compressor is coupled to a friction-type clutch and driven by a 1800-rpm shaft from the main gearbox. It features two internally driven sets of male/female rotors, each geared to the compressor's 1800-rpm output gear. Lubrication and gas path sealing are achieved via the compressor's lube oil system. No pump is required; the discharge to suction pressure gradient is used for flow metering. The fuel gas and oil mixture leaving the compressor enters a separator tank that removes all but 2 to 5 ppm of oil from the gas. A downstream coalescing filter reduces the oil carry-over to less than 1 ppm.

Gearbox

The gearbox was specifically designed to meet the single, prime-reliable drive system objective. Six shaft penetrations are provided which transmit power to and from the skid. In addition to the gas turbine and generator, the gearbox interfaces with the gas compressor, turbine starter, boiler feed pump and lube oil pump. A plan view of the gearbox is shown in Figure 7.

The gearbox has a 4850 kW (6500 hp) input rating at a 1.3 service factor. It has a double reduction gear set between the 14,200-rpm engine power shaft and the 1800-rpm bull gear. A 6400-rpm intermediate gear drives the boiler feed pump while a 5100-rpm intermediate gear is used for the hydraulic starter. The 1800-rpm shaft near the center of the gearbox drives the electric generator and lube oil pump. At an outboard location, another 1800-rpm shaft drives the fuel gas compressor.

Start-Up and Control

The integration of the mechanical drives within the load gearbox affects the start-up procedure. The system requires fuel oil for starting because the gas compressor is driven by the load gearbox.

An electric-driven hydraulic pump provides energy through the hydraulic starter to accelerate the engine to ignition speed, approximately 2200 rpm. The acceleration to ignition speed occurs with the gas compressor declutched. After light-off and subsequent declutching of the starter at 8500 rpm, the engine accelerates to its design speed of 14,200 rpm. At this time, the gas

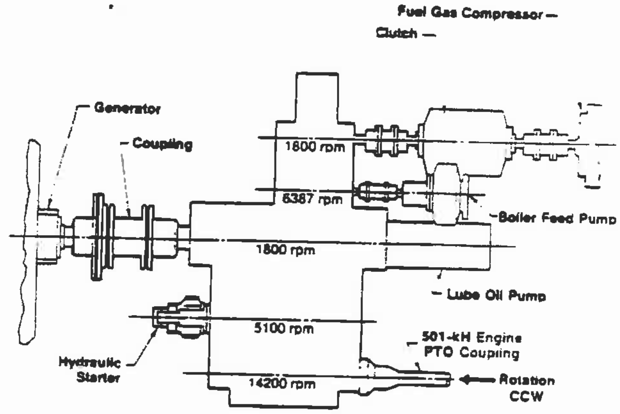

FIG. 7 SYSTEM GEARBOX - PLAN VIEW

compressor's clutch is engaged and the gas compressor begins producing high-pressure gas. At a fuel gas pressure of approximately 1930 kPa (265 psig), the engine's fuel control is automatically switched from oil to gas. No discernible change in speed or output is expected during fuel switching. The manner in which the system can readily accommodate either natural gas or distillate, both in the gas turbine and duct burner, is a major feature of this system.

During steady-state operation, the engine's firing rate is used to maintain steam pressure at the process header. The control logic responds to pressure changes in the following manner:

- Steam Pressure Decreases: Indicative of increased demand, the turbine firing rate (fuel flow) is increased to meet steam demand. If the maximum TIT is exceeded, the supplementary fired duct burner is operated until steam demand is met.
- Steam Pressure Increases: This occurs when there is an excess of steam being delivered to process. In conventional systems, this would result in a droop in the turbine's firing rate causing a simultaneous reduction in output and poor heat rate. In this case, however, the reduction in steam header pressure is effected by diverting excess steam into the turbine. Without changing the firing rate, the header pressure falls and generator output increases. However, if the generator output exceeds the new set point or maximum output (4000 kW_e), the turbine firing rate is then reduced accordingly.

CLOSURE

A packaged cogeneration system has been specifically developed to overcome the obstacles that have thus far delayed the widespread use of cogeneration in the industrial process sector. The totally integrated concept combined with the steam injection feature results in better performance, reliability, and reduced complexity. The successful demonstration of this system at a General Motors plant in early 1985 will confirm this approach and will set the standard for packaged cogeneration systems.

ACKNOWLEDGEMENTS

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