Review of a Combined Steam and Gas Turbine Cycle for Pipeline Service

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The combined steam and gas turbine cycle provides the highest efficiency turbine system available today. In view of the rapidly escalating value of fuel the combined cycle therefore merits a review for pipeline applications. Such a review reveals the combined cycle has a number of advantages. First, the combined cycle efficiency is significantly higher than the efficiency of a standard regenerative cycle gas turbine. Second, and contrary to the characteristics of a standard gas turbine, the efficiency at a given load improves significantly as the ambient temperature increases, so that the combined cycle would be applicable in hot climates. Third, the adjustable speed capability of the combined cycle meets the usual pipeline service requirements. This paper briefly presents the results of a preliminary study of a combined cycle single drive system as it might be utilized in a gas pipeline station.

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As most mechanical engineers know, the steam and gas turbine cycle concept is not new to industry, but has been under study and in use nearly as long as gas turbines. Open-cycle gas turbines discharge, a great amount of useful heat in the exhaust gas. For this reason, almost all gas turbines in industrial plants utilize this exhaust heat in one way or another.

The electric utility industry appears to be turning more and more to combined cycles for smaller size fossil-fueled power plants because of their high efficiency. If combined cycles will have a major role in the electric utility industry, would they have a place in the pipeline industry for the same basic reason? Engineers in the author's company have made a preliminary investigation of this question. This paper presents some of the findings.

There is a major difference in the use of combined cycles or electric power plants and for pipeline centrifugal compressor drive. Electric power generators operate at constant speed, but pipeline centrifugal compressors are called upon to operate at various speeds and loads. The adjustable speed capability of the combined cycle is one of the key areas of interest for pipeline service.

It should be noted that combined cycles are not new to the pipeline industry. Four combined cycle systems were installed on pipelines between 1968 and 1970 and are now operating successfully. In addition, there are two older stations which use gas turbine exhaust heat for separate steam turbine driven compressors. However, since these installations were conceived, gas turbine technology has advanced and system efficiencies have increased considerably. The rapidly escalating value of fuel indicates the more complex combined cycle should again be investigated.

**Cycle Description and Selection**

The cycle chosen for investigation is shown in Fig. 1. A standard simple-cycle two-shaft gas turbine, rated 14,600 hp at ISO conditions, exhausts its heat to an unfired heat-recovery steam generator. The steam generator is of a design well within current practice, and extracts what is initially thought to be an economically optimum amount of the gas turbine exhaust heat. The design selected for this review includes an economizer, evaporator, and superheater. It produces 54,500 lb/hr of 410 psig 793 F steam at ISO conditions.

The steam turbine is connected directly to the outboard side of the single centrifugal compressor to make a single drive system with a 100 percent rated speed of 6500 rpm. The combined ISO rating of this system is approximately 22,000 hp, with the gas turbine providing approximately 14,000 hp and the steam turbine approximately 8000 hp.

An alternate combined cycle system would use a regenerative cycle gas turbine with exhaust gases passing through both a regenerator and a heat-recovery steam generator. If the gas turbine were separately rated 13,750 hp at ISO conditions, this regenerative combined cycle would produce approximately 15,400 hp at ISO conditions.

Both of the foregoing combined cycle arrangements have a thermal-cycle efficiency of approximately 39.2 percent on a lower heating value basis.
at ISO conditions. This efficiency includes approximately 300 hp auxiliary losses and is four or five points better efficiency than available from the best gas turbine efficiencies offered today.

It is, of course, possible to use a combined cycle arrangement with the steam turbine driving a second separate compressor. Where the single compressor system is suitable from a station pressure-flow point of view, however, it would be more economical. The two compressor system installed cost would be higher because of the additional compressor, piping, valves, and controls.

Operation of the single compressor combined cycle system is basically simple. The steam turbine has no steam flow control valve, and generates power proportional to the steam delivered from the heat-recovery steam generator. Therefore, just as for a standard gas turbine, load control is accomplished by increasing or decreasing gas turbine fuel. The combined cycle pipeline station would, therefore, provide the desired station discharge pressure simply by proportional adjustment of the gas turbine fuel governor.

BASIS OF SYSTEM ANALYSIS

In order to have a basis for this preliminary evaluation, the combined cycle is compared with a modern standard gas turbine. Further, it is not sufficient to compare drive systems at rated efficiencies only, because on pipelines, these systems operate at various speeds and loads.

For our purposes, we assume the compressor station horsepower and pressure ratio vary with the flow rate as shown in Fig. 2. These characteristics are typical of what might be called normal compressor operating lines. They represent a series of steady-state operating points at constant station discharge pressure and different flow rates, and with the upstream station operating in similar fashion. The slopes of these two characteristics vary slightly with different line diameters, flow rates, etc., but all stations have characteristics similar to this.

Of course, there are also operating points that do not fall on these normal operating lines; and operation at these special conditions must also be evaluated; but for our purposes here, we will only use the normal operating lines.

For these normal operating conditions, a properly selected centrifugal compressor has a normal operating line as shown in Fig. 3. This horsepower-speed characteristic is important in order to determine the centrifugal compressor and gas turbine operating efficiencies. As pipeliners know, if the 100 percent horsepower and 100 percent speed point is at the compressor peak-efficiency point, then all of the points on the normal operating line down to a very low speed will be at relatively high efficiency. Also, this operating line falls in a very good range of regenerative-cycle gas turbine efficiencies, but the actual efficiencies depend significantly on ambient air temperature.
Fig. 5 Thermal efficiencies at various ambient temperatures, loads, and speeds, for a standard gas turbine and for a single-drive combined cycle. These operating lines are used as the basis of fuel calculations for the illustrations used in this paper.

COMBINED CYCLE EFFICIENCIES

Fig. 4 illustrates the estimated thermal efficiencies for the basic operating conditions of Figs. 2 and 3. The four solid lines represent thermal efficiencies of the 22,000 hp combined steam and gas turbine drive system at various steady-state compressor flows. Ambient air temperatures of 0, 30, 59, and 90 °F were selected to show the effect on thermal efficiency.

The upper right end of each curve represents the maximum flow and horsepower capability of the drive system at the indicated ambient temperature. The system was assumed to require fully rated ISO horsepower to produce 100 percent standard cubic feet compressor flow. Therefore, the upper limit of the 59 °F curve is 100 percent flow and rated 39.2 percent efficiency. As the required flow decreases, the required compressor drive horsepower and speed decrease exponentially as in Figs. 2 and 3, and the resultant thermal efficiencies are estimated for these off-design points.

The dashed lines represent a modern regenerative cycle standard gas turbine, with a rated efficiency of 34.3 percent. LHV when operating on natural gas fuel. These dashed lines, representing operation of the standard gas turbine at various flows again in accord with Figs. 2 and 3, are super-imposed on the combined cycle curves in order to allow visual comparison.

These curves illustrate the significant differences between the two types of turbine drives as they would operate on a pipeline. The dashed lines of the standard gas turbine show the well-known result of various ambient temperatures. At increasing ambient temperatures, the standard gas turbine efficiencies decrease significantly. On the other hand, for the combined cycle at a given flow and load, the efficiency is higher at higher ambient temperatures. At 90 °F and maximum drive capability, the combined cycle efficiency is approximately seven points (21 percent) better than the standard regenerative cycle gas turbine.

Another comparison of the two types of drives is shown in Fig. 5. For 100, 75, and 50 percent horsepower loads at the required reduced speeds for assumed pipeline operation, this illustration gives the estimated thermal efficiencies of the two types of turbine drives systems. The basic data used for Fig. 5 is the same as for Fig. 4. Again, Fig. 5 shows for a given constant load, the efficiency of the combined cycle improves as ambient temperature increases. With higher ambient temperatures, the steam flow increases and the
steam turbine power contribution increases without increase in gas turbine fuel.

The efficiency advantage of the combined cycle in hot weather is illustrated by Fig. 6. The assumptions which are the basis for Fig. 6 are:

1 Fuel is evaluated at $1.00 per million Btu.
2 The 22,000-hp ISO rated combined cycle is compared with a standard gas turbine rated 22,000 hp and 34.4 percent efficiency at ISO conditions.
3 The two drives are assumed to run at a steady load and at a constant ambient temperature for 8500 hr.

The upper line indicates the difference in fuel costs if the two drivers were to run steadily at their maximum horsepower capability for 8500 hr. For example, if the ambient temperature were 59 F for 8500 hr, and the two drivers ran steadily at their maximum capability of 22,000 hp for 8500 hr, fuel for the combined cycle would cost $169,000 less than fuel for the standard gas turbine. For a 29-year assumed life, and at an interest rate of 10 percent, the present worth of this annual savings is $1,583,000. And if the value of fuel continues to rise, and interest rates increase, the present worth would be made more.

The lower line in Fig. 6 illustrates the difference in fuel costs if the two drivers were to operate steadily at 75 percent of their ISO rating, and at 89.5 percent speed (as on a pipeline), for 8500 hr. Under these conditions, and below 30 F ambient temperature, the standard gas turbine uses less fuel, as would occur for operation in Alaska or northern Canada. At high ambient temperatures, however, the combined cycle fuel savings would be very significant.

OVERALL STATION ECONOMIC ESTIMATES

Obviously, the combined cycle drive system is more complex than the standard gas turbine system. One of the major requirements is cooling water for the steam condenser. Where cooling water is available, however, the added complexity may be justified.

Although this report is not intended to cover all the many possible pipeline situations where a combined cycle drive system might be considered, one estimate of total station costs is given here to indicate how the higher installed costs of the combined cycle pipeline station might be offset by the lower fuel costs. Refer to Fig. 7 which shows estimated installed and operating costs of a 22,000 hp combined cycle and a 22,000 hp standard gas turbine.

The fuel costs given in Fig. 7 are based on operation of the two drivers at their maximum horsepower capability at 75 F ambient, and at the speed indicated by Fig. 3. Operation was considered to be at these conditions constantly for 8500 hr, a condition that would not actually exist, but might be considered representative for a pipeline in a hot climate.

Maintenance costs vary with many factors, including the number of starts and stops a turbine undergoes. The costs given here are based on experience with both gas and steam turbine systems, and at less than ten start/stops per year. The major difference in the two systems is the maintenance cost of the steam generator and steam turbine, estimated to be a relatively minor amount in comparison with other costs.

The lube oil costs should be approximately the same for the two drive systems, but steam system water requirements increase the costs of the combined cycle over the standard gas turbine which has no water makeup requirements.

The total annual costs shown here indicate the combined cycle is the lowest, although the margin is not great. As the value of fuel increases in the future, and for operation in warm climates, the combined cycle should, therefore, be an increasingly viable alternate for large unit compressor drivers on pipelines.