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Summary of the Research
and Development Effort on
the Supercritical CO₂ Cycle

A. P. Fraas

Fossil
Energy
Program

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SUMMARY OF THE RESEARCH AND DEVELOPMENT
EFFORT ON THE SUPERCRITICAL CO₂ CYCLE

A. P. Fraas

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SUMMARY OF THE RESEARCH AND DEVELOPMENT EFFORT ON THE SUPERCRITICAL CO₂ CYCLE

A. P. Fraas

ABSTRACT

The supercritical CO₂ cycle has the advantage over a conventional closed-cycle gas turbine in that the compression work phase of the cycle can be carried out close to the critical point and hence aerodynamic losses in the compressor are reduced and the cycle efficiency increased for a given turbine inlet temperature. However, the practicable turbine inlet temperature is reduced by permissible stresses in the heater tubes because the peak pressure in the cycle must be ~260 atm in order to have the compression process take place close to the critical point of the working fluid. The high system pressure also makes the capital cost of the heat exchangers and gas piping higher than that for a conventional closed-cycle gas turbine. Further, the waste heat from the cycle must be rejected at too low a temperature for it to be useful for industrial process heat or for district heating systems.

INTRODUCTION

This is one of a series of topical reports each of which reviews the research and development (R&D) effort on one of a set of advanced energy conversion systems.^{1,2} The first portion of each report outlines the background by describing the system, indicating its principal advantages and problems, and briefly summarizing the research and development effort on the system up to 1976, including the cumulative costs and system operating times. The next section points up the vital problem areas and recommends experiments deemed most likely to resolve these problems. This is followed by a section giving a more detailed discussion of the principal problem areas, performance parameters, and figures of merit characteristic of the system. The next section gives the current status of the development work with particular reference to progress in improving the principal performance parameters. The last section reviews the principal experience in the R&D work on the system.

This work has been carried out at the request of the Office of Program Planning and Analysis of the DOE Fossil Energy Program using funds provided for a general appraisal of advanced fossil energy systems.

BACKGROUND

The first study of record on a supercritical CO₂ thermodynamic cycle was carried out by Samuels in 1958 for use with a gas-cooled reactor.³ This cycle is a special case of closed-cycle gas turbine systems that takes advantage of the low specific volume in the vicinity of the critical pressure and temperature to reduce the compressor work and hence make the cycle less sensitive to the aerodynamic efficiency of the compressor. In the 1958 studies it was found that the high system pressure required (about 260 bars) led to difficult problems in the design of the pressure vessels and piping, and this in turn led to high capital charges that more than offset the improvement in thermal efficiency obtainable.

In the 1960's, Feher proposed the supercritical CO₂ cycle for use in small power plants both for terrestrial use and for space nuclear electric power applications.⁴⁻⁷ A number of design studies were carried out, and a small system mockup was built and operated. Concurrently, a number of further studies were made of gas-cooled central station fission reactor plants coupled to supercritical CO₂ cycles.^{8, 9}

In recent years the cycle has been considered for use in fossil fuel-fired central stations on the premise that it would give a higher thermal efficiency than the closed-cycle gas turbine. Thus, it was one of the cycles considered in ECAS-I (Energy Conversion Alternatives Study, Phase I).⁹

System Description

Figure 1 shows a flowsheet for a supercritical CO₂ cycle designed to operate with a fluidized-bed coal combustion chamber as the heat source. The three fluidized-bed coal combustion systems at the left operate in parallel heating the CO₂ coming from the high-temperature recuperators. The hot, high-pressure CO₂ leaving the heat transfer

Fig. 1. Supercritical CO₂ cycle (Ref. 9).

matrices in the fluidized beds flows to two turbines in series, the first of which drives the pump and compressor and the second the generator. Note that both a pump and a compressor are used in conjunction with the low-temperature recuperators and precoolers to give a little better thermodynamic cycle efficiency than if a pump only were employed.

A temperature-enthalpy diagram for the cycle is shown in Fig. 2. The critical pressure and temperature for CO_2 are 73 bars and 31°C , respectively (1072 psia and 88°F). To stay close to the critical point without getting into condensation and the extra problems that this would entail, the minimum pressure must be kept somewhat above 73 bars and, for a reasonable pressure ratio for the cycle, it is necessary to go to a peak pressure of the order of 260 bars (3780 psia). The strength of iron-chrome-nickel alloys suitable for the heater falls off so rapidly at temperatures above 600°C that the peak cycle temperature must be limited to about 732°C (1350°F) because of creep strength considerations. Thus, there is little flexibility in the choice of cycle operating conditions.

Note in Fig. 2 that a great deal of heat must be added in the recuperator system between the compressor outlet (point 2 in Fig. 2) and the heater inlet (point 3). In fact, this amount of heat is of the order of three times the heat added in the heater, and the heat addition must be at a small temperature difference if the large amount of heat available in cooling the turbine exhaust (the path from point 5 to point 6 of Fig. 2) is to be employed to good advantage in obtaining a high overall cycle efficiency.

Advantages and Disadvantages

The prime advantage of the supercritical CO_2 system is that it gives a cycle much less sensitive to aerodynamic losses in the compressor than is the case for conventional gas turbines. It also reduces the size of the components, but this is done at the expense inherent in the design and construction of pressure vessels and piping for high system pressures. Header sheets, pressure vessel penetrations, and flanged joints become particularly awkward to handle, and disproportionally high costs result. (Note that the pressure involved is a little higher than that commonly

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Fig. 2. Temperature-entropy diagram for an ideal supercritical CO_2 cycle. (Ref. 4).

used in supercritical pressure steam plants, and the temperatures are substantially higher so that either substantially greater wall thicknesses or a complex double wall with intermediate thermal insulation must be employed.) Because of these limitations, the peak practicable temperature in the cycle cannot be as high as in a closed-cycle gas turbine that would operate at a pressure 25% or less than that required in the supercritical CO_2 system. Thus, the increase in the practicable peak temperature for a conventional closed-cycle gas turbine in large measure offsets the reduced losses in the compressor power requirements of the supercritical CO_2 system.

The advantages of the supercritical CO_2 cycle stem from careful selection of the key pressures and temperatures in the cycle. As a consequence, operation at reduced loads entails a substantial loss in thermal efficiency, whereas in a closed-cycle gas turbine the thermal efficiency remains substantially constant from full load down to around 25% load, an important advantage in a utility system because loads vary substantially throughout the day.

Both open- and closed-cycle gas turbines reject their heat at a fairly high temperature, so that they are well suited to applications requiring the cogeneration of heat and electricity. However, the supercritical CO_2 cycle rejects its heat at a relatively low temperature, 64°C (149°F) and below, and hence is not suited for applications requiring both electricity and heat at moderate temperatures.

The supercritical CO_2 system has the further disadvantage that the turbine and compressor required would be of special designs different from any equipment currently on the market. There is no question that suitable units could be designed and their compact size should reduce costs once they were in production, but, as with the helium closed-cycle gas turbine, a potential barrier to the construction of an experimental plant lies in the special character of the machinery required. This, of course, is not the case for open or closed cycles employing air as the working fluid.

Operating Experience

The only supercritical CO₂ system that has been built was a reduced scale mockup of a 250-kW(e) system designed for the Army Corps of Engineers and built by McDonnell-Douglas. This 15-kW(t) heat input system did not include a turbine generator; a throttling valve was employed to simulate that component. The system was operated for 25 hr.

Summary of Costs and Running Time

A total of about \$2,500,000 has been spent on design studies for a supercritical CO₂ system. Of this, about \$500,000 was funded by the U.S. Army Corps of Engineers and the balance was private funding by McDonnell-Douglas. In addition, between \$50,000 and \$100,000 was spent in developing a reference design for the ECAS-I study, and about the same amount was spent in the latter 1950's in the course of studies of advanced gas-cooled reactor systems. As indicated above, the only running time on the supercritical CO₂ system was on a small mockup that was operated for about 25 hr.

RECOMMENDATIONS

There is no active program on the supercritical CO₂ system at the time of writing, and a review of its advantages and problems indicates that the likelihood of an important payoff from the development of this system does not appear to be sufficient to justify initiation of a new program.

PRINCIPAL PROBLEMS AND PARAMETERS

The principal problems and parameters of the supercritical CO₂ system are much the same as those for the closed-cycle gas turbine, and the reader is referred to a companion report for a discussion of these items.¹⁰ For example, the fireside corrosion problem in a fluidized-bed furnace is essentially the same for both systems as are the other problems of fluidized-bed coal combustion systems. Similarly, such parameters as

the ratio of pumping power to heat removal, recuperator effectiveness, and the relative magnitude of capital and fuel costs are much the same. Thus, this section is concerned with just two areas in which there are major differences, that is, the choice of working fluid and the capital cost of the recuperator.

Choice of Working Fluid

If one wishes to take advantage of the reduced work of compression for a closed-cycle gas turbine system that operates with the compression portion of the cycle close to the critical pressure and temperature, one finds that the fluids of Table 1 have been considered as promising candidates. Note that water has been included, although in practice it has been found best to operate it on a Rankine cycle so that the pump work is carried out in the liquid phase and thus is even less than if it were in the gas phase. The fluorocarbons, SO_2 , and SF_6 do not appear to be good candidates because they would tend to break down or dissociate at high temperatures and give serious problems with corrosion. These problems could be avoided with xenon, but this material is very expensive

Table 1. Critical constants of working fluids⁶

Name	Formula	Critical temperature (°F)	Critical pressure (psia)
Ammonia	NH_3	271.2	1636
Carbon dioxide	CO_2	87.8	1072
Hexafluorobenzene	C_6F_6	460.0	402
Perfluoropropane	C_3F_8	161.4	388
Sulfur dioxide	SO_2	315.5	1143
Sulfur hexafluoride	SF_6	114.0	546
Water	H_2O	705.0	3206
Xenon	Xe	61.9	853

and has properties little different from CO_2 . Ammonia could be employed, but both its critical temperature and its critical pressure are higher than for CO_2 , making CO_2 the more promising candidate. Thus, CO_2 seems to be the best candidate of any of the fluids that have been considered for a supercritical pressure cycle in which there would be no condensation and no problems with two-phase flow such as instabilities and difficulties in controlling free liquid surfaces. (This was one of the advantages of the supercritical CO_2 cycle for space power plant applications where control of free liquid surfaces under zero- g conditions poses major problems.)

Capital Costs

Capital costs are difficult to estimate even for modest variations in conventional systems. Inasmuch as most of the systems of interest here involve major developmental uncertainties, it has seemed best to consider only the effects of the design conditions inherent in roughly similar systems on the relative cost of the most expensive components (that is, the high-temperature heat exchangers). Extensive experience indicates that the cost of these units is approximately proportional to both the surface area required and the pressure if the operating pressure is above about 1.0 bars. Further, the cost increases rapidly with the design temperature for temperatures above about 500°C , the point at which the strength of low-alloy steel begins to fall off rapidly. More expensive alloys must be considered as the design temperature is increased above 500°C because of both creep-strength and fireside corrosion considerations (see Table 2).¹¹

With the above considerations in mind, it is interesting to consider the design temperature and pressure conditions for the high-temperature heat exchangers required for typical cycles together with the temperature difference available, the heat flux, and the ratio of the heat energy that must be transferred to the electrical output. The latter is a particularly important factor for the closed-cycle gas turbine and the supercritical CO_2 cycle because both depend on recuperators in which the amount of energy transferred is two to four times the useful power

Table 2. Allowable stress, tube wall thickness, and cost for 0.75-in.-OD tubing of typical alloys for 3500 psi internal pressure and temperatures above 200°F

Tube wall temperature (°F)	ASME allowable stress (Psi)	Tube wall thickness for 3500 psi (in.)		Tube ID* (in.)	Tube cost (\$/ft)	Tubing cost (\$/ft ² internal surface area)
		No allowance for corrosion	0.050 in. allowance for corrosion			
<u>Type 304 SS</u>						
1200	6,000	0.120	0.170	0.410	2.50	23.30
1350	2,900	0.178	0.228	0.294	3.10	40.28
1500	1,400	0.232	0.282	0.186	4.20	86.25
<u>Hastelloy X</u>						
1200	11,300	0.077	0.127	0.496	19.00	146.32
1350	4,800	0.137	0.187	0.376	26.00	264.13
1500	3,000	0.175	0.225	0.300	33.00	420.00
1650	1,200	0.242	0.292	0.166	47.00	1081.00
<u>Incoloy 800</u>						
1200	7,200	0.106	0.156	0.433	3.80	33.14
1350	3,700	0.158	0.208	0.334	5.00	57.18
1500	2,000	0.207	0.257	0.235	6.20	100.00

*Including 0.050 in. additional thickness to allow for corrosion.

output of the cycle. Values for some typical cases designed on the basis of similar criteria were taken from the author's files to yield the numbers in the first five columns of Table 3, and this made it possible to compute the sixth column. Data on tubing cost as a function of temperature and wall thickness¹¹ then gave a basis for calculating the relative costs given in the extreme right-hand column.

The relative cost values appear to be consistent with a number of previous studies and provide some insight into the reasons for differences between systems. For example, the heat transfer matrix costs for the closed-cycle gas turbine of Table 3 are roughly half again as high as for the steam plant. The still higher cost for the supercritical CO₂ cycle is consistent with the ECAS-I results, which for the same cycle pressure and temperature conditions as used in Table 3 yielded heat exchanger costs six times as high for the supercritical CO₂ system as for the helium closed-cycle gas turbine. Almost half of this stems from the much greater amount of heat that must be transferred in the recuperator per unit of electrical output, and the balance apparently stems from the thicker tubes and the awkward header design problems associated with the very high pressures inherently required in the supercritical CO₂ cycle.

In discussing the above capital cost problems with E. G. Feher (the leading advocate of the supercritical CO₂ cycle), the writer learned that, in 1975, Feher and his colleagues at the Actron Division of McDonnell-Douglas found that they could reduce the cost of the recuperator by a large factor by using small-diameter tubing arranged in tube bundles with relatively small header sheets and then coupling these with a larger header sheet within a large shell housing the complete assembly. This approach was used by Escher-Wyss for closed-cycle gas turbines in the 1940's¹² and certainly should give lower costs than estimated by General Electric in ECAS-I⁹ for conventional shell-and-tube units. However, Escher-Wyss experience indicates that it still did not make the closed-cycle gas turbine competitive with steam systems, and Table 3 indicates that the capital cost of the recuperator for the conventional closed-cycle gas turbine is inherently much lower than for the supercritical CO₂ cycle. Note, too, that the NASA review of the ECAS-I work also viewed the

Table 3. Comparison of factors affecting the size and cost of the high-temperature heat transfer matrices for some typical power plants

	Cycle peak pressure (bars)	Cycle peak temp. [°C (°F)]	Mean temp. diff. (°C)	Ratio of heat transferred to electric output	Average Heat flux [W/cm ² (Btu/hr-ft ²)]	Relative surface area	Temp.- cost factor	Relative cost
Pulverized coal-fired steam plant	246	565 (1050)	~400 (720)	2.5	1.26 (3,900)	2.5	1.0	2.5
Fluidized-bed steam plant	246	565 (1050)	~400 (720)	2.5	3.15 ^a (10,000)	1.0	1.0	1.0
Closed-cycle gas turbine								
Heater	58	815 (1500)	167 (300)	2.5	3.15 (10,000)	1.0	2.0	0.6 ^b
Recuperator	68	538 (1000)	56 (100)	2.1	0.63 (2,000)	4.2	0.8	<u>0.9</u> 1.5
Supercritical CO ₂								
Heater	260	733 (1350)	250 (450)	2.5	4.4 (14,000)	0.7	2.0	1.5
Recuperator	260	594 (1100)	56 (100)	4.6	1.9 (6,000)	3.1	0.8	<u>2.6</u> 4.1

^aMinimum wall thickness requirements when allowing for fireside corrosion cause the tube wall thickness and cost to vary non-linearly and to be independent of pressure for pressures less than ~10 bars.

^bAverage heat flux for both the boiler tubes in the fluidized bed and superheater and reheater tubes in the hot gas stream leaving the bed.

General Electric cost estimate for the recuperator in the supercritical CO₂ cycle as too high, and their efforts to evolve a less expensive design did yield a lower estimated cost. However, this reduced cost was still not low enough to make the system really attractive.¹³

REFERENCES

1. A. P. Fraas, *Comparison of Advanced Energy Conversion Systems on the Basis of R&D Experience*, ORNL/TM-6259 (to be published).
2. A. P. Fraas and A. S. Thompson, *R&D Experience and Current Status of Development of Open-Cycle Gas Turbines*, ORNL/TM-6251 (to be published).
3. M. Bender et al., "Comparative Features of Gas-Cooled Fast Reactors," Reactor Study, Oak Ridge National Laboratory, 1959.
4. E. G. Feher et al., *System Analysis and Design Concept of a 150 KWE Supercritical Thermodynamic Cycle Power Conversion Module*, DAC-60998-F (January 1969).
5. E. G. Feher et al., *Investigation of Supercritical (Feher) Cycle*, AFAPL-TR-68-100 (October 1968).
6. E. G. Feher, "The Supercritical Thermodynamic Power Cycle," paper presented at the 1967 Intersociety Energy Conversion Engineering Conference, Miami Beach, Fla., Aug. 13-17, 1967.
7. J. R. Hoffman and E. G. Feher, "150 KWE Supercritical Closed Cycle System," *Trans. ASME*, Vol. 93 (1971).
8. E. Maillet and M. Raveau, "Use of Carbon Dioxide Turbines for Advanced Reactors," IAEA Symposium on Advanced and Gas-Cooled High-Temperature Reactors, Jülich, Federal Republic of Germany, Oct. 21-23, 1968.
9. D. H. Brown et al., *Energy Conversion Alternatives Study (ECAS), General Electric Phase I Final Report, Volume II, Advanced Energy Conversion Systems, Part 2, Closed Turbine Cycles*, NASA CR-134948, SRD-76-011 (February 1976).
10. A. P. Fraas, *Summary of the R&D Effort on Closed-Cycle Gas Turbines*, ORNL/TM-6237 (to be published).
11. A. P. Fraas, "Heat Exchangers for High Temperature Thermodynamic Cycles," ASME Paper No. 75-WA/HT-102, Aug. 6, 1975.
12. C. Keller, "The Escher-Wyss Closed Cycle Turbine," *ASME Trans.* 68, 791 (1946).

13. *Comparative Evaluation of Phase I Results from the Energy Conversion Alternatives Study (ECAS)*, prepared by NASA Lewis Research Center for ERDA and NSF, NASA TM X-71855, pp. 145-48 (February 1976).

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