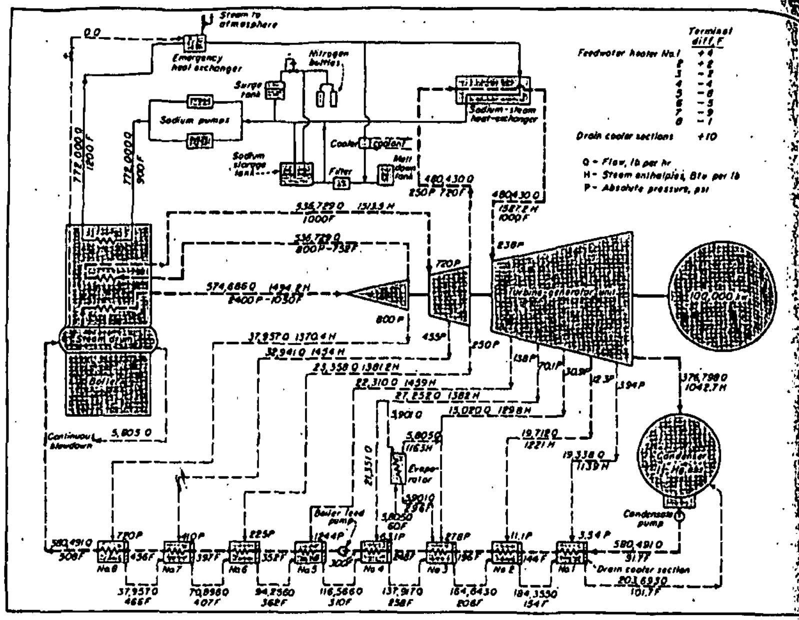

PLAN D, DOUBLE REHEAT CYCLE uses the liquid metal, sodium, as heat carrier to resuperheat steam at the second reheat point in the turbine circuit. Despite the increased investment added equipment this cycle may prove a practical possibility

Double Reheat Cycles—the Next Step?

Rising fuel costs and metallurgical limitations led the swing to reheat cycles. In the drive for more economy the double reheat cycles offer encouraging future possibilities

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► THE REHEAT CYCLE now used in power plants (1) increases plant thermal efficiency (2) reduces turbine maintenance by lessening moisture in exhaust steam. Using two stages of reheating, instead of one as at present, offers practical possibilities of further raising plant thermal efficiency.

Instead of piping steam back to the boiler for reheating, a heat-carrying fluid may be used. A liquid metal like sodium may circulate between a heater in the boiler gas pass and an exchanger near the turbine where steam emerges for reheating. With the sodium-steam heat exchanger near the turbine it is practical to design for a steam-pressure drop of 5% of reheat pressure leaving the turbine.

Study. In the table, opposite page, it has been assumed that normal reheating in the boiler causes a pressure drop of 10% of reheat pressure, while the heat-exchanger scheme involves only 5% steam-pressure drop. This accounts for the difference in station heat rates for Plan *A* vs *B*, and Plan *C* vs *D* vs *E*. All the plans are based on a 100,000-kw unit with eight stages of feedwater heating. Heat balance for Plan *D* is given in the circuit diagram above. Note that Plan E would use the liquid sodium system at both reheat points.

The sodium circuit includes: sodium-steam heat exchanger, pumps, piping and valves, surge and storage tanks, purifying equipment, gas-pressurizing apparatus, system heating units, equipment for filling system. In addition, an emergency heat exchanger allows the sodium system to dissipate heat during transient periods of startup and shutdown and when turbine steam supply suddenly ceases.

Comparison of Alternate Schemes of Single and Double Reheating Cycles

100,000-Kw Turbine-Generator With Eight Stages of Feed Heating

Plan	A	B	C	D	E
Throttle-steam conditions:					
Pressure, psia	2400	2400	2400	2400	2400
Temperature, F	1050	1050	1050	1050	1050
Turbine-inlet steam conditions, first reheat point:					
Pressure, psia	405	427	720	720	760
Temperature, F	1000	1000	1000	1000	1000
Method of reheating	Boiler	Sodium-stm heat exchanger	Boiler	Boiler	Sodium-stm heat exchanger
Turbine-inlet steam conditions, second reheat point:					
Pressure, psia	—	—	225	238	238
Temperature, F	—	—	1000	1000	1000
Method of reheating	—	—	Boiler	Sodium-stm heat exchanger	Sodium-stm heat exchanger
Condenser pressure, in. Hg abs	1.5	1.5	1.5	1.5	1.5
Boiler feedwater temperature, F	508	508	508	508	508
Turbine-generator unit heat rate, Btu per kwhr	7,502	7,462	7,293	7,256	7,216
Boiler efficiency, %	88	88	88	88	88
Auxiliary power demand, kw	7,000	7,000	7,000	7,000	7,000
Estimated net station heat rate, Btu per kwhr	9,167	9,118	8,911	8,866	8,817
Reduction in net station heat rate compared to Plan A, Btu per kwhr	0	49	256	301	350
Net station output, kw	93,000	93,000	93,000	93,000	93,000
Load factor, %	70	70	70	70	70
Operating hours per year	8,760	8,760	8,760	8,760	8,760
Estimated annual dollar saving compared to Plan A	0	$11,200	$58,300	$68,600	$79,800
Capitalized annual savings at 15%	0	$74,500	$388,000	$457,000	$531,000

For the single-reheat cycles, *A* and *B*, turbine section efficiencies were assumed at 85% for first section and 90% for the second. For the double-reheat cycles *C*, *D* and *E*, the first section was assumed at 85%, the second at 90%, and the third at 92%. A 1% mechanical and electrical loss was used for all cases. Feedwater temperature was assumed at 508 F for all plans.

While there would actually be a small difference in auxiliary power needs for each plan, the demand was assumed as 7000 kw for all. Turbine-shaft gland and valve-steam leakage were taken as zero throughout. Water compression effect on feedwater was neglected for all cases, not affecting the comparisons.

Liquid Sodium System. In the diagram, opposite page, sodium enters the boiler heating section at 900 F and emerges at 1200 F. Sodium has favorable characteristics as a heat-transfer fluid. Local heat-transfer coefficients between tube wall and sodium, equal to 5000 Btu per hr sq ft per F or greater, are easily achieved. Little difficulty should be experienced in limiting boiler-tube-metal temperatures to 100 F above the liquid-sodium temperature.

Centrifugal or electromagnetic-type pumps could be used for moving the molten sodium. From the boiler the 1200-F sodium flows to the steam resuperheater, a simple single-pass tube-and-shell counterflow unit. For this particular application where the sodium-steam heat exchanger is designed for a steam-pressure drop of 5 psi, the unit would be about 4 ft in diameter and 20 ft long.

The two pumps would each have a capacity of about 800,000 lb per hr with 100-ft tdh. This provides 100% standby capacity. For electromagnetic pumps about 100 hp at full rating would be needed. Centrifugal pumps need 80 hp or less. All piping and equipment in the sodium system for superheated steam reheating *would be stainless steel.*

Since sodium reacts with oxygen, a nitrogen pressurizing blanketing system is needed. The sodium-system purifying, filling and heating equipment are needed only for initial startups or after prolonged shutdown.

Estimated installed cost of the liquid-sodium system, including cost of the initial sodium charge, is about $300,000.

Economic Evaluation. Corresponding to the heat rates shown in table above, each cycle has the following thermal efficiency: *A*—37.2%, *B*—37.45%, *C*—38.3%, *D*—38.5%, *E*—38.7%. Table gives the reduction in heat rate for each plan or cycle referred to *A* as the base. These reductions were evaluated as follows:

1. Estimated load factor — 70%
2. Fuel cost — 40c per million Btu
3. Fixed charges covering amortization, interest, taxes, insurance — 15%
4. Period hours per year corresponding to load factor of 70% — 8,760
5. *Estimated annual dollar saving* — (Btu per kwhr reduction) (net kw station output) (hr per year) (load factor) (fuel cost per Btu)
6. Capitalized value of saving — (Estimated annual dollar saving)/0.15

Example

As an example the saving for Plan *B* over *A* is 49 Btu per kwhr, then the annual dollar saving — $49 \times 93{,}000 \times 8760 \times 0.7 \times 0.4/10^6$ — $11,200.

Then capitalized value of saving — $11,200/0.15 — $74,500.

The more-developed plans seem to indicate an adequate margin for any additional costs that might be involved by a slightly more complex turbine structure for the second reheat point. This cycle might be a practical answer to what can be done to fight inflationary fuel costs, while steam temperatures are limited.