heater costs are reflected in the type of pumping scheme chosen. This is necessary because excessive heater costs *would reduce some of the savings that* result from using superpressures.

DESIGN PROBLEMS

Performance. Feed pumps for superpressure plants require considerably more input horsepower *per pound of* steam delivered to main turbine than is required at lower pressures. It is, therefore, imperative that optimum pump *efficiency* be obtained in order not to jeopardize the advantages of the superpressure cycle.

Design of the high-speed pump for the various arrangements in Fig. 1A-1D *requires new approaches* to determine performance of the individual impellers and the correct shape of the discharge volutes. Tip speed is about 50% greater than that ordinarily used in modern pump practice. Our firm is developing basic design information by tests that check the performance of a complete stage operating at high tip speeds. Test rig is arranged to handle hot water so *actual field inlet conditions* can be obtained and the necessary net positive suction head for various capacities measured.

Compressibility. Some interesting problems present themselves when we make performance calculations for high-pressure pumps. We ordinarily think of water as being incompressible at the pressures encountered in usual boiler feed-pump practice. Actually, of course, water is compressible, and for pressures of 5000 psi this is definitely a factor to be considered.

Fig. 6A shows the rectangular pressure-volume diagram for an incompressible fluid. Diagram area indicates the *amount of work in foot-pounds* per pound that must be done to compress the fluid from suction to discharge pressure.

If the real properties of water are considered, the pressure-volume diagram *changes to the shape in Fig. 6B. This* shows that the theoretical work required is reduced somewhat by the volume change, V_1 to V_2. To illustrate the magnitude of the compressibility, assume that water at 500 F is compressed from 2000 to 5000 psi. Under these conditions the specific volume will be reduced 1.7%, assuming isentropic compression.

Because *work is done on the water* between points 1 and 2, Fig. 6B, the water temperature increases, even though pump efficiency is assumed to be 100%. For the 3000-psi pressure *rise discussed above, water temperature* will increase 10 F. Actual temperature rise will be still greater, depending on the internal losses in the pump. It is evident that it is necessary to consider compressibility during design of these pumps.

Shaft seal will be an extremely important part of a successful installation of a superpressure pump. Ordinary watercooled packing, as used in many *present designs, will not be suitable for* the pressures and speeds involved. This is particularly true of the secondary unit in Fig. 1C and 1D, where seal is exposed to 2100 psi and either 350- or 500-F water.

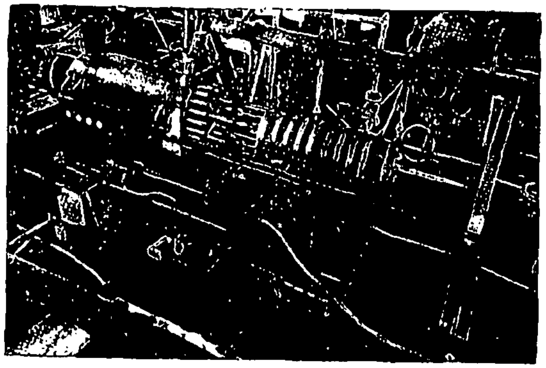
Design of the pump incorporates a packingless shaft seal. While this will require some leakage from the pump, a design is being developed that will keep this leakage to a minimum. *Work is* progressing on a test rig, Fig. 2, to check seal performance under actual operating conditions and to investigate the best materials combination suitable for the seal.

Materials. Research studies over a *number of years* have developed the proper chemical analysis of materials for feed pumps. Temperatures involved in superpressure applications are within the range of previous practice, requiring the use of chrome steels throughout the *pump interior.*

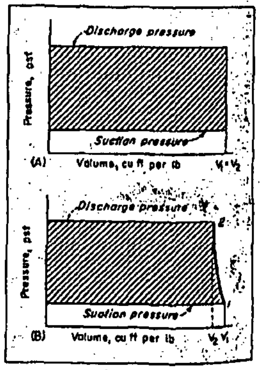
Physical-analysis studies indicate that wall thickness and stresses are well *within safe limits* and present no manufacturing problems. Some particular care will have to be taken to assure a good joint at the flange of the outer barrel of the high-pressure pump. *Preliminary investigations show that* a satisfactorily tight joint can be obtained with bolted construction, thus retaining ease of maintenance.

Our design includes provisions for bringing the inside and outside of the barrel up to operating temperature as uniformly as possible. Also, thermal expansion is provided for to minimize thermal stresses. Particular care must be given to expansion problems in high-pressure pumps *for* superpressure service.

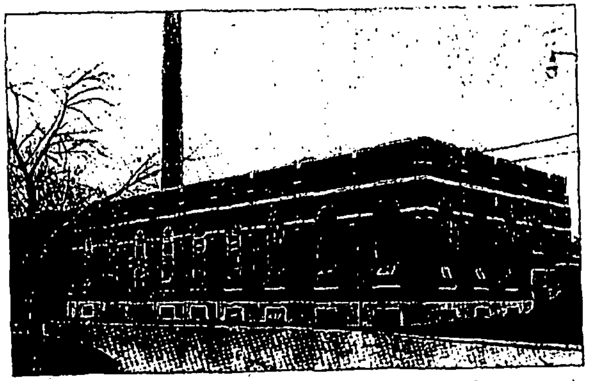
Maintenance. Reliability is of prime importance in superpressure plants. The barrel-type feed pump has proven itself to be well suited for this service. Maintenance, *when needed,* is *simple, can* be performed quickly, Fig. 5.



5 Barrel-type feed pumps require little maintenance, but when work must be done on a unit it is important that disassembly be easy. "A" frame here is big help



6 Pressure-volume diagrams for A, an incompressible fluid; B compressible



MUNICIPAL power-plant building has expanded twice to provide space for more capacity

Grand Haven Thrives With More Diesels

Installing slow-speed heavy-fuel-oil diesels successively is keeping this Michigan municipal plant out front in today's lively operating competition

By J BRYAN SIMS, *Superintendent, Board of Public Works, Grand Haven, Mich.*

►FROM THE INCEPTION of the municipal diesel power plant in 1931, we have consistently used large slow-speed diesel engines, burning straight-run residual fuel. This policy has paid off in fuel economy, low operating and maintenance costs.

In 23 years, plant capacity has grown from 1600 to 16,430 kw, keeping pace with load growth of the community. Grand Haven, which now has a population of 10,000 to 12,000, supports 51 industries that make a wide range of products.

Like many other municipal plants, we began with reciprocating steam engines, added turbines, and later turned to internal-combustion engines as a more efficient source of power.

Diesel Installations. The first two diesels installed in 1931 were DeLa-*Vergne 4-cycle 200-rpm 1140-bhp* 800-kw 6-cyl units. A similar unit was installed in 1934. In 1937, when the plant was expanded for two additional engines, a Nordberg 2250-bhp 225-rpm mechanical-injection 1600-kw 2-cycle 6-cyl unit was put on the line. In 1942, we added the fifth engine, a Nordberg 2-cycle 225-rpm air-injection diesel, rated at 3850 bhp, 2715 kw, 225 rpm.

In 1946, management, recognizing the need for additional power, studied the feasibility of extending and modernizing the 150-psi steam plant that had been retained for standby service. Coal could be purchased for $8 or $9 a ton. But the nature and size of the load was against economical steam operation, especially in view of the wide variance in load demand. *Besides, operating both* steam and diesel plants would have required additional personnel, since the steam plant is in a separate building.

So the diesel plant was extended a second time. The sixth engine to go on the line, another Nordberg 3850-bhp air-injection unit, started operation in June 1948. Total plant capacity rose to 9430 kw, but with peak loads zooming past the 7000-kw mark, the plant still did not have sufficient firm capacity.

The 1947-48 expansion had provided additional space for 4000 to 5000 kw in diesel power, or 5000 to 6000 kw in gas-turbine power. We took bids on both types of prime movers and made an investigation and analysis.

Advantages appeared for both types of prime movers. But over-all evidence indicated that the diesel engine was more advantageous for Grand Haven, and a 2-cycle, 9-cyl, 29x40-in. mechanical-injection diesel, rated 7000 bhp, 5000 kw net at 171.5 rpm, was ordered from Nordberg Manufacturing Co.

Before this unit could be built and installed, an anticipated power emergency required extra capacity at the earliest possible date. In addition, management wanted to save banking *costs* for fuel burned under the boiler in the steam plant. In June 1951, we installed two 1000-kw General Motors' surplus engines in the powerhouse basement. These engines operate at 720 rpm, burning a gas oil *or* No. 3 distillate. They supply standby and peaking power.

We also ordered a *Nordberg* 3000-kw 10-cyl 4300-bhp 225-rpm 21½x31-in. 2-cycle mechanical-injection unit, which will replace the No. 3 DeLaVergne engine when it is installed in late 1953.

The 5000-kw unit was placed in service April 11, 1952, giving the plant a total capacity of 16,430 kw. All the engines, except the two General Motors units, use residual oil for fuel.

Auxiliary Equipment. Table bottom of page 77 lists auxiliary equipment for the new 5000-kw engine. One of the interesting features of the installation