RESEARCH ACCOUNT

The history of liquid metal pump development in India

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In a fast breeder reactor, the liquid sodium coolant is circulated through the core using vertical centrifugal pumps. These pumps are critical equipment for reactor operation and they have many unique design features. Indigenous development of pumps is vital for the demonstration and expansion of fast reactor technology which constitutes the critical second stage of our three-stage programme.

Indigenous design and development of vertical centrifugal sodium pumps was taken up more than two decades ago in close collaboration with Indian industry. This article provides an account of the work done in the areas of pump hydraulics, mechanical design and testing, and manufacture.

Keywords: Cavitation, liquid metal, model testing, manufacture, sodium centrifugal pump.

In a fast neutron reactor, such as the prototype fast breeder reactor (PFBR) (Figure 1) which is under commissioning by Bharatiya Nabhikiya Vidyut Nigam (BHAVANI) at Kalpakkam, liquid sodium is used to transfer the heat generated in the nuclear reactor core to water in the steam generators to produce steam. The generated steam at high pressure (170 bar) and temperature (495°C) drives a conventional steam turbine to produce electrical power (500 MW). Liquid metal sodium is the preferred coolant in fast reactors due to its excellent heat-transfer characteristics and non-moderating property. This permits effective breeding of fissionable plutonium while producing (electricity) power. Circulation of liquid sodium in both the primary and secondary circuits (see note 1) is achieved through centrifugal pumps which are preferred over electromagnetic pumps because of their high efficiency and robust construction.

When conceptual design of the PFBR was launched in the mid-1980s, at the Indira Gandhi Centre for Atomic Research (IGCAR), the design envisaged four centrifugal sodium pumps each in both primary and secondary coolant circuits. Early efforts were directed towards developing a primary coolant pump of 7600 m^3/h capacity at 84 m pump head at an operating speed of 720 rpm and requiring a motor power of 2100 kW; the pump was to operate in radioactive environment at a temperature of 400°C. Following a study carried out to optimize reactor cost, the current design consisting of two pumps each in primary and secondary sodium circuits was finalized. This resulted in doubling of pump capacity and posed new challenges in its hydraulic development. Indigenous development of reactor sodium pumps and other critical fast reactor components, such as steam generators, was taken up early on in the fast reactor programme in response to the international embargo on sale of equipment to India for nuclear applications. (While initially dialogues were held with pump suppliers such as KSB Germany and also with SERENA/FASTEC combination of the Western European nations for technology transfer, these were never taken further because of embargo considerations. This was especially true of liquid sodium pumps which differed substantially from other reactor pumps such as pressurized heavy water reactor (PHWR) coolant pumps that have been long available from KSB Pumps (India) supported by KSB Germany.) The development of indigenous technology was therefore vital in realizing a self-reliant programme. Although major Indian suppliers were contacted, it was soon realized that no pump industry could assist us fully in this mammoth maiden effort because of complex technological requirements, such as high operating temperature, radioactive environment, modest Net Positive Suction Head (NPSH), in sodium guide bearings, etc.

This article therefore traces the history of reactor pump development in India, including the challenges that were overcome in the areas of hydraulics and manufacturing, which have now made indigenous manufacture of such pumps a reality.

Background and early developments

The interest in sodium pump development at IGCAR started with the commissioning of an imported centrifugal pump installed in a sodium heat transfer loop in the then Reactor Engineering Laboratory (REL). During its procurement, the pump did not have the benefit of third-party inspection, nor was it water-tested as claimed by the supplier (water has similar fluid flow properties as sodium). Even the assembly/dismantling procedures had to be formulated using a simple pump assembly sketch. The

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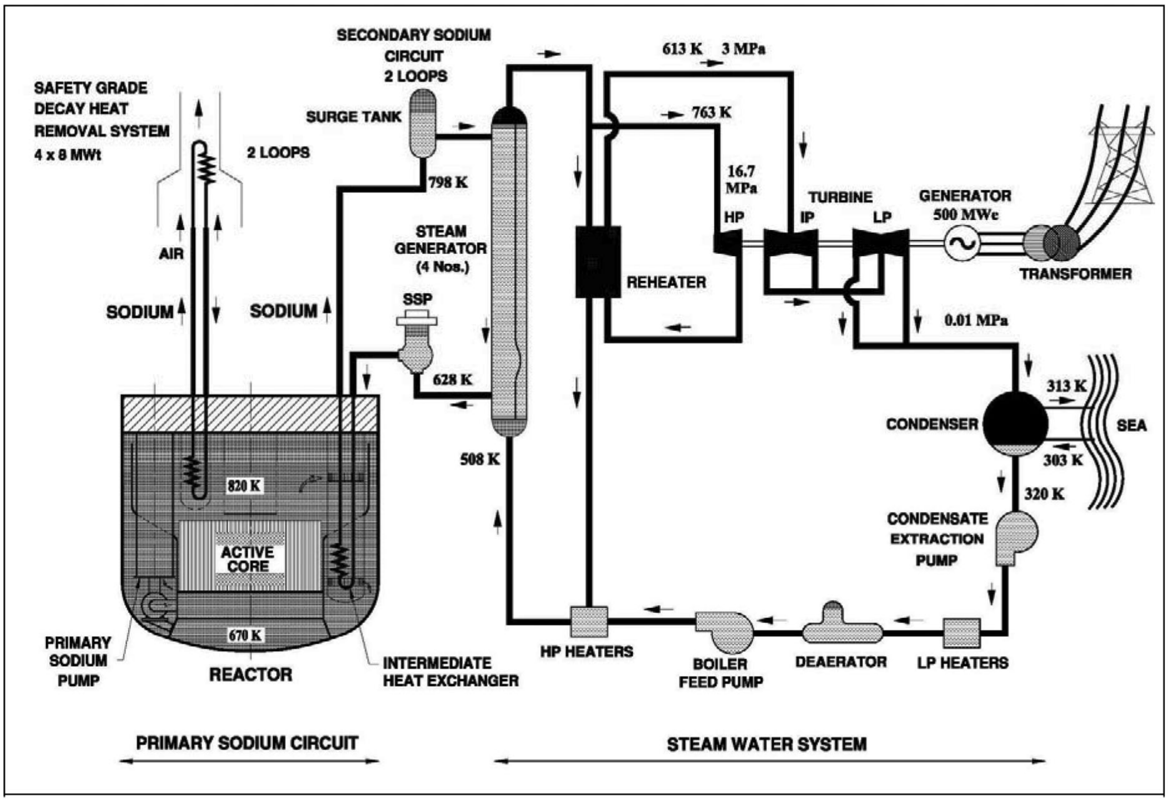

Figure 1. Schematic of prototype fast breeder reactor.

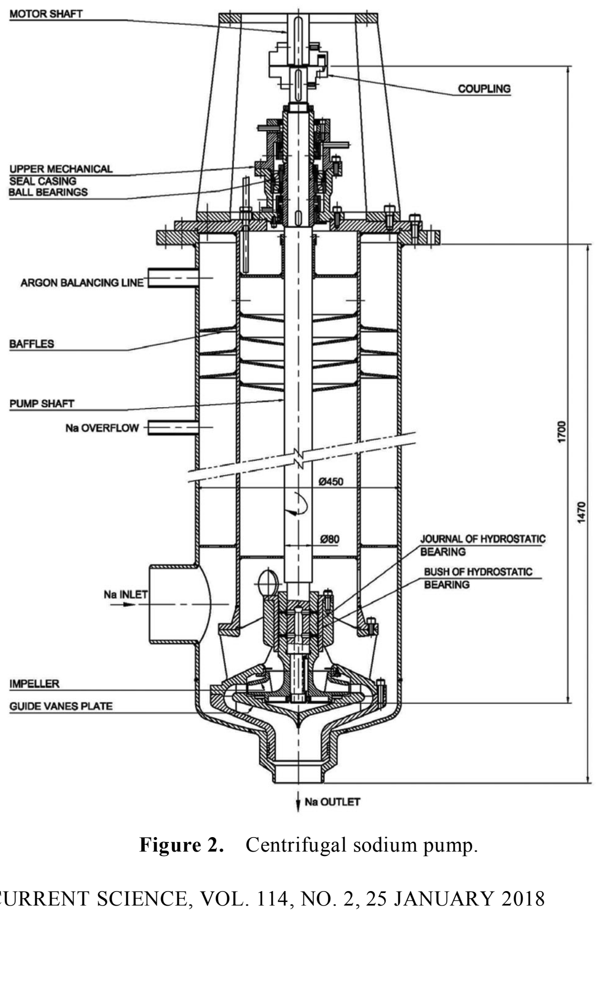

Figure 2. Centrifugal sodium pump.

water-testing and later sodium commissioning of this pump brought into the open several problems, particularly related to its sodium lubricated hydrostatic bearing, and the concomitant troubleshooting exercise provided the much needed exposure in the areas of bearing design and pump construction.

A small experimental centrifugal sodium pump (50 m^3/h, 7 kW, 2900 rpm speed) was later built to gain insight into sodium pump design and construction[1] (Figure 2). The material of construction was stainless steel, except for shaft seals and upper thrust bearings. All hydraulic parts were made in radiographic quality CF 8 castings from a reputed supplier in South India, whereas all welding and satellite coating on hydraulic bearing parts were performed by another fabricator under departmental guidance/supervision. The pump was first water-tested in a specially built loop and its hydraulic performance, including NPSH tests were satisfactorily performed. Later, the pump was tested at its design temperature of 375°C in sodium in the existing engineering sodium loop. Figure 3 shows hydraulic performance in water and sodium. The pump was run in sodium over a flow range of 10–43 m^3/h (ref. 2). The H–Q characteristics in sodium were found to be lower than those in water tests, which was contrary to expectations. The shortfall in performance was attributed mainly to increased internal leakage flow resulting from poor fit between discharge nozzle and

Table 1. Problems encountered in fast reactor pumps

Reactor	Problems encountered
EBR-II[25]	Both primary pumps seized after operating for less than 200 h during initial testing. This resulted in galling of bush of lower bearing.
FERMI[26]	Water hammer problem experienced with check valve at pump discharge.
	Wearing of mechanical seal faces.
	Drive thrust bearing replacement.
	Rheostat in speed control system repaired.
	Eddy current coupling repaired.
RAPSODIE[27]	Seizure of primary and secondary pumps during initial isothermal tests.
BOR 60[28]	Leakage of seal cooling coil of secondary sodium pump and ingress into sodium.
KNK II[29,30]	Bowing of shaft from annealing of residual stresses resulting in increased shaft residual unbalance and vibration.
BN-350[31,32]	Hydraulic shock experienced during closure of check valve at pump discharge.
	Cavitation damage of primary and secondary coolant pump impellers.
PHENIX[3]	Failure of check valve during commissioning.
	Loosening of bush of hydrostatic bearing of both primary and secondary pumps.
PFR[4,5]	Oil leak into primary sodium.
	Hard facing (stellite) of hydrostatic bearing of secondary pump detached.
	Shaft bowing/seizure in primary pump.
BN-600[6,32]	Cavitation damage of primary pump impeller.
	Damage of teeth of coupling.
FFTF[7]	Bowing of shaft of pump.
SPX-1[8]	Oil leak in the drive unit of primary sodium pump.
FBTR	Minor problems due to speed regulation, oil leakage in the upper seal box and presence of loose parts in secondary sodium pump.

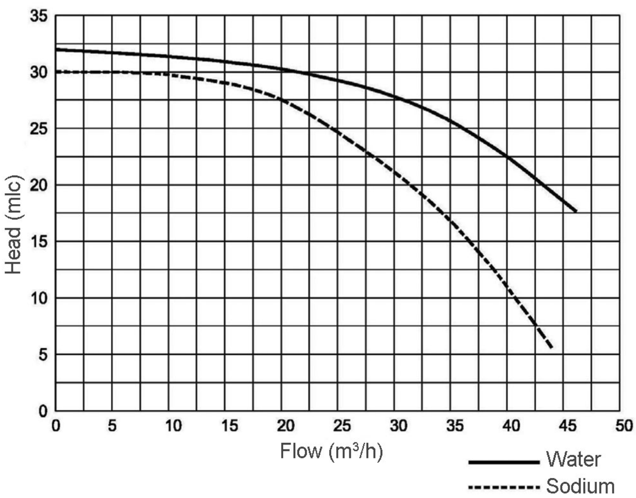

Figure 3. Pump hydraulic performance in water and sodium.

outer casing of the existing pump in sodium heat transfer loop and increased clearance of 280 μm in the hydrostatic bearing compared to 200 μm during water-testing. The pump was operated in sodium for 300 h. The vibration and noise measurements compared reasonably well with those recorded in water tests.

The experience of building and testing the small-capacity pump did provide some insights into pump hydraulic design. The approach, however, was theoretical and based on simple textbook formulae, unlike those of reputed pump manufacturers which are based on empirical data accumulated over years of testing different designs.

The effort, however, generated confidence that in-house R&D efforts coupled with the experience of Indian pump industry could be synergized to realize indigenous pump design for high-temperature sodium applications.

Experience with reactor centrifugal pumps

Sodium centrifugal pumps necessarily operate in a high-temperature, radioactive environment with rather modest NPSH. While the operating temperature can be lowered by choosing the option of locating the pump in the cold leg (a choice available only for loop-type reactors), the NPSH available cannot be increased significantly because this will necessitate the use of high cover gas pressure or large shaft span between bearings, neither of which is preferred due to the resulting difficulty in sealing the active cover gas or lowering of pump rotor critical speed.

Early reactor pump experience in several countries revealed problems of shaft bowing and seizure, distortion of bottom hydrostatic bearing support, check valve malfunctioning and even cavitation damage[3–8].

Table 1 provides a summary of problems experienced in fast reactor pumps. The experience gained highlights the need for careful design and technology development coupled with thorough testing to ensure satisfactory operation of reactor pumps.

Development of reactor primary pump (version I)

The initial pump development programme focused on developing the reactor pump by subassemblies or parts and later integrating the same. Having completed the design and construction of the small-capacity pump described above, the manufacturing technology development of the full-scale primary pump was undertaken. This included a long shaft of composite geometry (partly hollow and partly solid), impeller castings, and hardfacing of bottom hydrostatic bearing parts. A conceptual design of the pump taking into consideration constraints on size, possible intake and discharge arrangements, impeller and diffuser configuration was prepared. This was useful in determining representative sizes of components and finalizing the manufacturing technology. Simultaneously, a technical dialogue was initiated with several reputed Indian pump suppliers as well as some foreign suppliers for evolving a firm hydraulic pump design through a two-stage process. The first stage involved developing a 1/3 scale model pump, and the second stage involved construction and performance testing in water of a full-scale 'hydraulic prototype' alone. The contract was finalized by competitive bidding and tender evaluation through a two-step process consisting of initial evaluation of technical competency followed by price evaluation and selection of the lowest bidder.

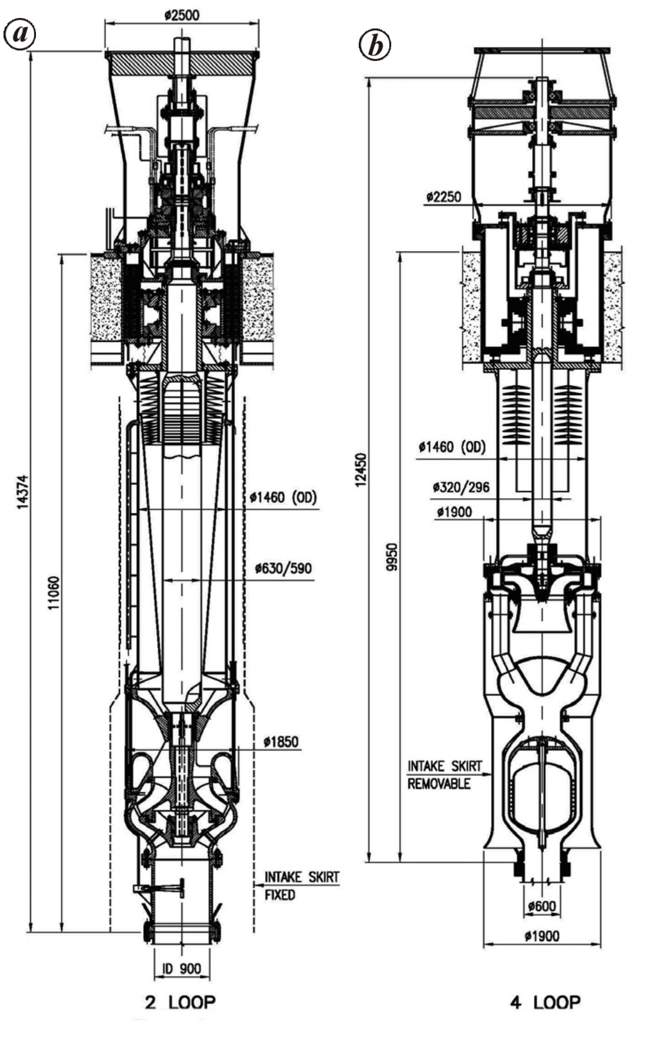

Figure 4. Comparison of primary sodium pumps.

Primary sodium pump (version I) – description and specifications

The primary sodium pump was a single-stage pump with bottom suction impeller and radial diffuser arrangement (Figure 4 *a*). The adopted hydraulic layout allowed smooth fluid entry into the impeller eye without turning the flow, thus minimizing entry losses. This also provided higher submergence yielding a high NPSH. A similar concept was also adopted in several reactor pumps built elsewhere. The pump was of vertical construction with overall height of approximately 12.5 m and maximum barrel diameter of 1.9 m. The free sodium surface in the pump barrel was topped by argon inert gas. Leak tightness of the argon gas space from the reactor building was achieved by a double mechanical seal around the pump shaft. The shaft of approx. 7.2 m length was guided in sodium by a bottom hydrostatic bearing which received pressurized feed of sodium from the impeller discharge. An operating speed of 700 rpm was selected so as to achieve a reasonably compact pump without compromising on the NPSH margin, thereby eliminating the possibility of cavitation in operation.

The following important specifications were to be achieved by the supplier:

1. A capacity of 7600 m^3/h at 82 m pump head at a bowl efficiency (see note 2) of 85%.
2. Limiting the overall size of the removable hydraulic parts to 1900 mm with an operating speed close to 700 rpm. This restriction was important because the lateral dimension of the pump removable part had a direct bearing on the reactor vessel diameter and therefore the reactor capital cost.
3. Optimizing the impeller geometry to achieve a suction specific speed (see note 3). This corresponded to a margin of 1.8 on NPSH available (NPSHA) (i.e. $NPSHR_{3\%}$ = NPSHA/1.8).

In addition, the hydraulic design was to be validated through model tests wherein an additional requirement of complete absence of visual cavitation in the impeller inlet area was to be demonstrated. It was considered sufficient to do model tests in water, as the hydrodynamic properties of water at room temperature and those of sodium at the reactor operating temperature of 397°C (670 K) are similar (Table 2).

Model tests

A 1/3 scale geometrically similar model was selected in conformance with the stipulations of the Hydraulic

Institute Standards (HIS)[9], which mandate that (i) the model pump head is at least 80% of the prototype so as to ensure adequate turbulent flow in the model, and (ii) the model impeller outlet diameter is greater than or equal to 300 mm. A model testing speed of 1900 rpm was selected based on pump affinity laws

$$N_{\mathrm{m}} = N_{\mathrm{p}}\sqrt{\frac{H_{\mathrm{m}}}{H_{\mathrm{p}}}} \times \frac{D_{\mathrm{p}}}{D_{\mathrm{m}}},$$

where subscripts m and p refer to model and prototype respectively.

A speed of 1900 rpm was preferred over the standard motor speed of 1450 rpm, as it limited the brake horse power at the shaft while satisfying the HIS requirements.

A closed-circuit test rig consisting of a water tank (with free surface level) and 300 mm piping was erected and the model pump installed in end suction configuration. The tank was provided with a facility to regulate air pressure in the tank and thereby vary the NPSH available at the pump inlet. This feature also facilitated degassing the system to minimize the effect of dissolved air on cavitation performance. The tank was also provided with a facility to maintain the temperature of water in the circuit below 30°C using a 60 tonne refrigeration plant. This was necessary to minimize the thermodynamic effect of temperature on cavitation performance[10].

Additional arrangements for visual cavitation study: To facilitate visual observation of incipient cavitation, a portion of the suction pipe adjacent to the model pump, the suction casing and the impeller front shroud were made of transparent material (Perspex) (Figure 5). During testing, the impeller vanes were observed under stroboscopic [model B&K 4912] lighting for the presence of vapour cavities during the cavitation tests.

The experimental procedure consisted of first lowering the NPSH from an initial large value wherein no cavitation existed, until significant vapour cavity patches were observed in all vane passages and then increasing the NPSH in small steps of 0.3 to 0.5 m until the cavity patches disappeared in all the vanes (five vanes). The NPSH at which the cavity patches disappeared in most vanes was taken as visual inception point, $\mathrm{NPSH}_{\mathrm{vis}}$. These tests were performed at 80%, 100% and 120% of the rated flow at the test speed of 1900 rpm. Figure 6 shows the model test results.

Conclusion of model tests[11]: The results from the model test were used to predict prototype pump performance at the design speed of 700 rpm. It was observed that the estimated prototype pump head was higher than the expected value by 7%. This necessitated either lowering of speed or trimming of the impeller outlet diameter (less than 3%), if required. The measured $\mathrm{NPSHR}_{3\%}$ on the model pump was 5.5 m, which indicated an exceedingly high suction specific speed of 248 (Q (m^3/s) H (m) and N (rpm)). The bowl efficiency was found to be 88% against the specified minimum of 85%. The pump performance was considered satisfactory and the experience gained in the process of designing and testing a pump of high suction specific speed was valuable.

Testing of hydraulic prototype[12]

Despite satisfactory results on the model pump, prototype testing was considered necessary to achieve the following:

(a) Eliminate uncertainty in cavitation performance ($\mathrm{NPSH}_{\mathrm{R}}$ value), especially because of the high suction speed design.

Table 2. Comparison of fluid dynamics properties of water and sodium

Fluid units	Water	Water	Sodium[33]
Temperature (K)	293	313	670
Density (kg/m^3)	998	992	856
Kinematic viscosity (m^2/s*10^7)	10.03	6.58	3.26
Surface tension (N/m)	0.0736	0.0701	0.1667

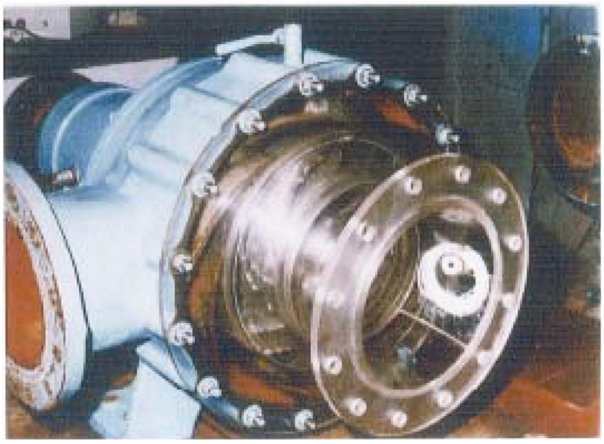

Figure 5. Model for visual cavitation testing.

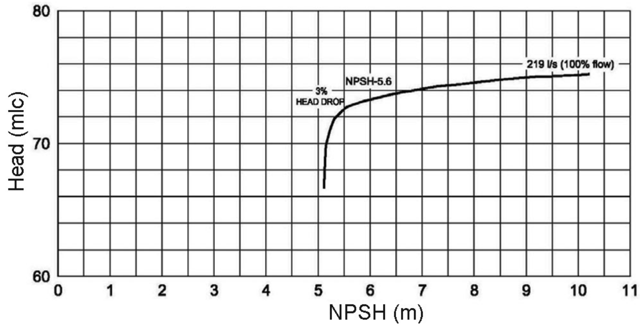

Figure 6. Cavitation performance of four-loop model pump.

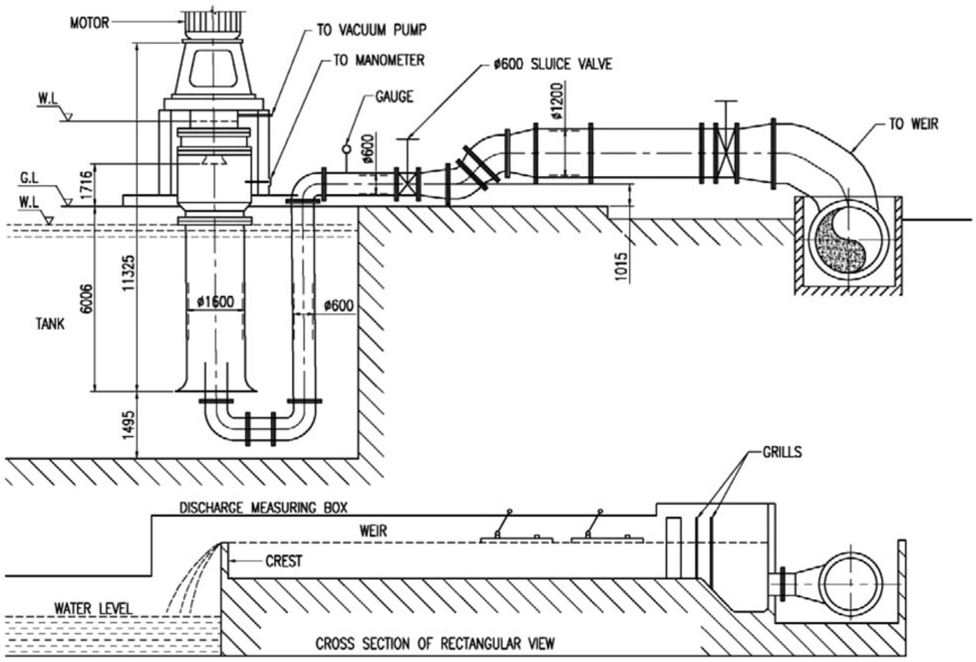

Figure 7. Test arrangement for hydraulic prototype.

Table 3. Comparison of results from prototype testing with those extrapolated from model tests

Parameter	Model test at 1900 rpm	Model extrapolated to 600 rpm (trimmed impeller)	600 rpm test results (trimmed impeller)
Flow (m^3/h)	806	6788	6788
Head (mlc)	73	63.9	64.4
Efficiency (%)	84	88*	86
Power (kW)	195	1352	1385
$NPSHR_{3\%}$ (mlc)	5.5	>5.04#	4.74

*Efficiency has been extrapolated for the untrimmed impeller.
#$NPSHR_{3\%}$ has been extrapolated for the untrimmed impeller.

(b) Eliminate uncertainty arising from transferring results from a horizontal model (for ease of visual detection of cavity) to a vertical pump, primarily from changes in the intake path.
(c) To determine the effect, if any, of impeller trim on $NPSH_R$ (refs 13, 14).

The hydraulic prototype, instead of a complete prototype pump, was employed to limit the cost incurred in these additional tests. For the same reason, the material of construction was carbon steel for all parts, except the impeller and diffuser which alone were made from austenitic stainless steel.

The rated speed of 720 rpm could not be employed because of power supply limitations at the supplier's test bed. A lower speed of 600 rpm was selected while simultaneously ensuring that this reduced speed did not exceed the lower limit prescribed by HIS (according to HIS, the lowest test speed is 80% of operating speed)[9].

Figure 7 shows the test rig arrangement for the hydraulic prototype in the manufacturer's open sump test facility. The open sump arrangement required that the impeller be raised above sump water level by a minimum of 1.76 m so as to achieve sub-atmospheric pressure at the pump suction during NPSH tests. Priming of the pump was achieved using vacuum pumps connected to the pump suction/casing.

For varying NPSH during cavitation tests, the HIS prescribed method of regulating air pressure above the free surface level at the pump suction could not be employed as system was open to atmosphere. Instead, liquid level in the open pump was varied and the break-off point from the *H–Q* characteristic monitored to identify the $NPSHR_{3\%}$. Table 3 shows the performance test results.

The maximum efficiency obtained was 86% compared to the value of 88% predicted from model tests. The marginal reduction in efficiency was attributed to trimming of the impeller. However, the measured value of $NPSH_{3\%}$ at 600 rpm was lower than the predicted value (i.e. 4.74 m instead of the predicted value of 5.04 m from the model tests). The values of Thoma's cavitation coefficient for measured parameters in model and prototype were in close agreement, viz. $\sigma_m = 0.076$ and $\sigma_p = 0.074$ respectively.

Hydrostatic bearing test

After completion of all hydraulic performance tests, the hydraulic prototype was dismantled and the bottom sleeve bearing replaced by a hydrostatic bearing designed at IGCAR. Concentricity of the journal with the bearing was checked during assembly and the pump was operated for 8 h. After completion of tests the pump was disassembled and visually inspected to confirm absence of significant rubbing/scratch marks on the bearing surface.

Pump manufacturing technology[15]

The primary pump was a large vertical assembly of approximately 12 m in height. The rotor shaft itself was 7.2 m in length and of composite geometry (partly solid and partly hollow) with a maximum diameter of 0.30 m. The impeller casting diameter was ~1.2 m and weighed ~600 kg. The objective of the technology development exercise was to achieve self-reliance in the fabrication of large, intricate pump components.

Development of pump castings

These included mainly the impeller, the discharge casing and the bottom bearing support. These castings, being always immersed in high-temperature sodium, were to be of high radiographic quality standards to ensure long life. The radiographic quality grade-2, specified according to ASTM standard, excluded all defects in categories D, E, F and G corresponding to cracks, hot tears, etc. while permitting porosity and inclusions to a limited extent. In order to meet this requirement, it was necessary to ensure high quality at the casting pouring stage itself. This was possible only through the use of high-quality zircon sand for both mould and cores, in view of its high temperature stability and better cooling characteristics.

Two impellers were cast, viz. a small impeller of 400 mm in diameter for the model tests and a full size impeller of 1270 mm diameter for the second stage testing on the hydraulic prototype. They were made in austenitic stainless steel CF8, and both castings satisfied the chemical composition and mechanical properties requirements.

In the case of the smaller casting, ASTM level-2 radiographic quality level was achieved. During the manufacture of the larger casting, additional care was taken to coat the cores with zircon powder in order to improve the as cast surface finish. The casting was radiographed extensively using Ir192 source. The radiographs showed unacceptable defects in a few locations at the blade to shroud junction, which was accepted considering the size and complexity of the component. Figures 8 and 9 show the mould assembly for casting the impeller and the large impeller casting during machining respectively.

Manufacture of pump shaft

The primary pump (version I) had a 7.1 m long shaft of composite geometry, viz. a central tubular section welded at both ends to massive solid sections. This particular geometry was necessary to achieve sufficient margin between the critical speed of the rotor assembly and the pump operating speed, while maintaining a reasonable shaft weight (Figure 10). The need to operate with close running clearances at the bottom and top bearings separated by a span of 5400 mm, and the limit on concentricity of 0.02 mm between various diameters and the reference diameter required a high standard of manufacture (e.g. distortion-free welding, and 'bright' stress relieving of the shaft assembly).

Welding and heat treatment of shaft

A joint design with no root gap and a backing-cum-centring ring was used for the two circumferential welds

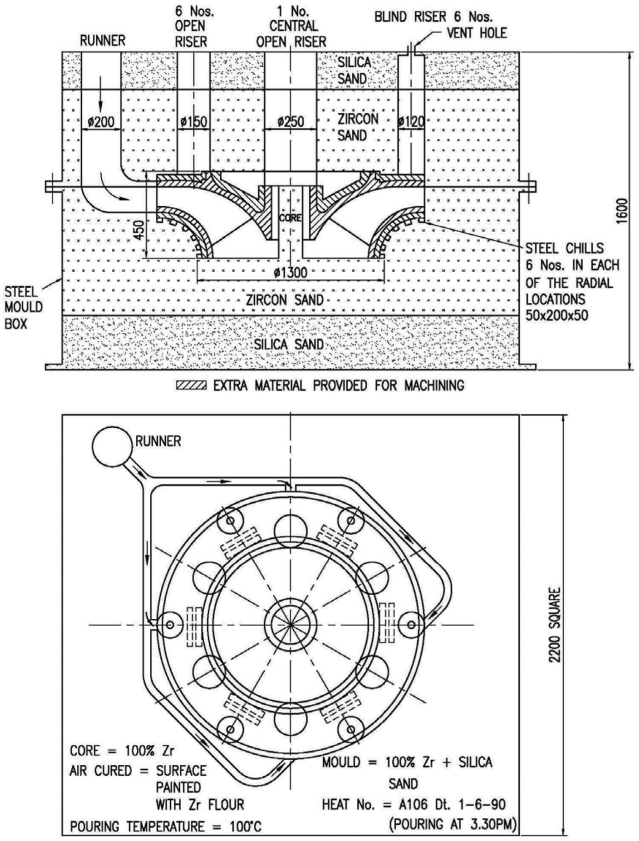

Figure 8. Mould assembly for casting of large SS impeller.

between the hollow and solid portions of the shaft; this helped in minimizing shrinkage and distortion. Also, distortion during welding was continuously monitored using dial gauges and corrections were implemented after each welding pass by appropriate welding sequence. The experience during welding suggested that stage radiography is to be carried during welding, one after root pass and another after 60% weld completion.

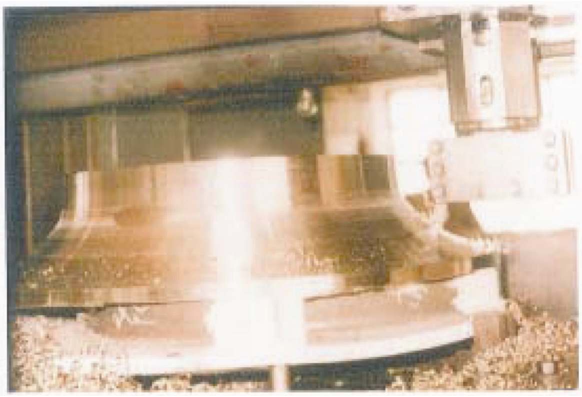

Figure 9. Photograph of large casting.

The shaft was then post-weld heat-treated to relieve all welding and machining-induced stresses so as to ensure dimensional stability during long operations at high temperature (400°C). Figure 11 shows a temporary retort and furnace used for the purpose. Inert atmosphere of argon + 10% hydrogen was maintained in the retort in order to retain the surface finish/brightness of the shaft, as no machining was permitted on the hollow portion afterwards. The finish machining of the solid portions was carried out with low depth of cut and feed to avoid inducing stresses in the shaft. Dynamic balancing of the shaft assembly was carried out to limit the mass unbalance to 838 g-cm, which represents high grade of balancing (ISO-grade G 2.5 or better) as in gas turbines. Figure 12 shows a photograph of the shaft at the IGCAR site.

Development of compact sodium pump – background

Much before the completion of testing of hydraulic prototype, the IGCAR management constituted a committee, including senior scientists from Bhabha Atomic Research Centre, Mumbai to explore alternatives in order to reduce

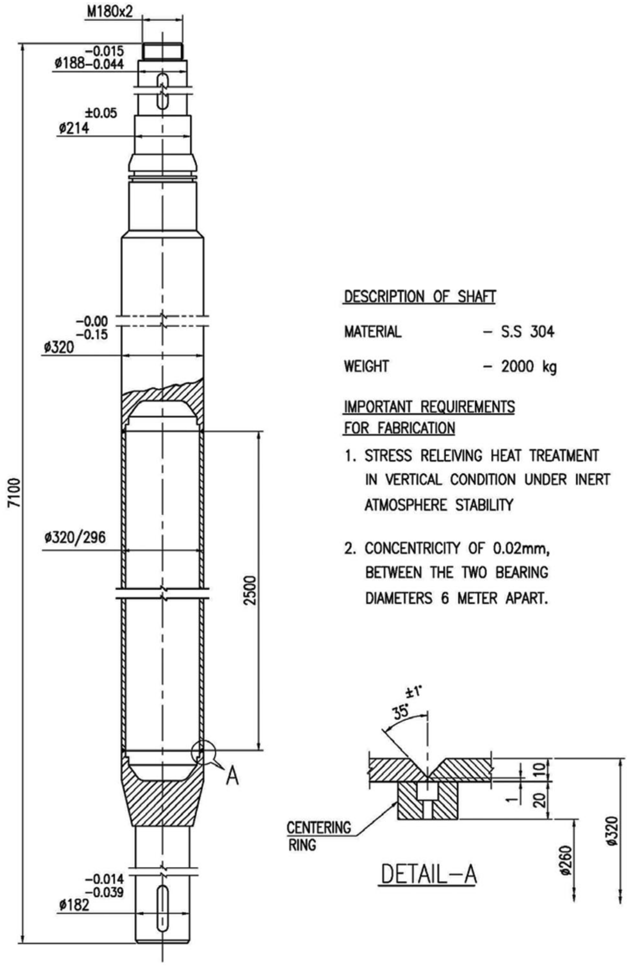

Figure 10. Schematic of prototype pump shaft.

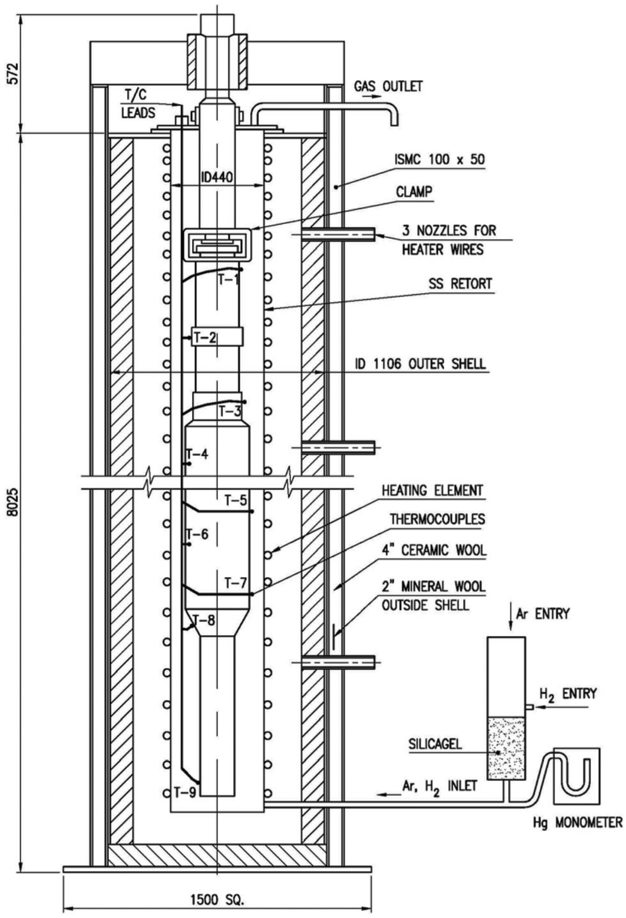

Figure 11. Stress-relieving heat treatment.

both the capital cost and construction time of the 500 MWe reactor. The emphasis was to reduce the number of systems and components (by scaling up component capacities) as was being pursued for the demo/commercial fast reactor plants in Western Europe and other nations such as Japan. The committee recommended use of the following:

- Two main coolant pumps (primary pumps) and four intermediate heat exchangers in the primary circuit.
- Single suction primary pump as in the French fast reactor plants of 250 and 1250 MWe.
- Two secondary heat transport loops with one pump in each loop and a total of eight steam generators (four in each loop), to minimize reduction in operating power in case of failure of one steam generator.

By mid-1990s, preliminary design work had been undertaken for a primary pump with double (15,000 m^3/h) the capacity of the previous design. Simultaneously, a fresh technical dialogue was initiated with the erstwhile supplier for designing a compact hydraulic assembly; this was limited to developing the pump hydraulics through model study alone. It was decided to limit the size of this double-capacity pump by adopting a reduced cavitation (NPSH) margin while retaining a single suction design. Detailed cavitation testing on the model pump, such as visual cavitation and cavitation erosion studies, was taken up to guarantee against cavitation damage. These detailed tests were considered necessary because the revised cavitation criterion adopted made it necessary to operate with limited cavitation while not significantly affecting impeller life. The revised cavitation criterion was selected to ensure that the pump size remained unaffected, although there was a doubling of pump capacity. It may be noted that pump size directly affects the main vessel diameter and has a direct bearing on the reactor capital cost. The technical dialogue with pump industry took considerable time for finalization as the erstwhile supplier (and also other pump suppliers) was not familiar with this newly stipulated cavitation criteria of erosion. The supplier had to develop a model to suit a full-scale pump of 4.3 m^3/s under a plant NPSH of ~15.5 to 16 m at a head of 75 m.

Figure 12. Photograph of prototype pump shaft.

Conceptual design of compact primary pump

While the dialogue for hydraulic development progressed with the erstwhile supplier during the mid-1990s, the conceptual design of primary pump evolved in-house. Additional research and development required to validate hydraulic and mechanical design concepts, envisaged in the new design, was also formulated.

The design finalized was a top suction, single-stage layout with suction flow turning through 180° to enter the impeller. The pump was to deliver flow rate of 4.3 m^3/s at a head of 75 m under a plant NPSH of ~15.5 to 16 m. The discharge flow was routed through a semi-axial diffuser downward into the discharge pipe leading to the reactor inlet manifold (Figure 4 *b*). This design was chosen over the bottom suction design used for the four-loops concept, because it simplified the delivery pipe layout (see note 4). The shaft was supported at the bottom (in sodium) by a hydrostatic bearing located above the impeller and at the top by conventional thrust/radial bearings, and the cover gas in the pump was sealed from the atmosphere by triple mechanical seals.

Figure 4 shows a comparison of the compact primary pump with the design made for four-loop primary circuit pump. The diameter of the removable pump assembly was kept well within 1900 mm, which was the diameter of the four-loop design pump having half the capacity of the compact primary pump. This was possible by increasing submergence (through elimination of non return valve (NRV) present in the earlier four-loop design) and more importantly, due to adoption of revised cavitation criterion (which was eventually proved to be adequate through model tests). The pump shaft diameter, however, had to be significantly increased due to longer span between bearings as well as due to significant increase in pump torque. Table 4 shows a comparison of the new pump with the proposed and operating large capacity pumps.

Table 4. Primary sodium pump data (large capacity pumps)

Parameter/reactor plant	BN 800 Russia*	BN 600M Russia†	SPX II France†	EFR Europe	PFBR India**	DFBR Japan†	SPX-1*
Capacity (m^3/h)	12,300	17,000	19,500	29,650	15,500	11,460	16,000
Delivery head (m)	101	70	65	71	75	95	75
Speed (rpm)	980	740	711	531	590	855	500
Specific speed (rpm) $\sqrt{\frac{m^3}{s}} \Big/ m^{\frac{3}{4}}$	56.85	66.35	72.28	62.28	47.93	50	41
Power (kW)	4000	3500	6200	–	3600	3400	4800
Efficiency (%)	72	80	-	–	82	–	80
Cover gas pressure (MPa)	0.055	0.055	0.01	–	0.01	–	0.007
$NPSH_A$ (m)	18.5	20	16	19	15.5	20	16
Speed variation range (%)	25–100	20–100	20–105	–	20–100	–	25–100
Sodium temperature (max; °C)	500	400	390	–	400	395	395
Total weight (tonnes)	120	95	72	–	–	–	120
Weight of removables (tonnes)	67.5	46	41	–	46	–	91
Hydraulic layout	SSDS	SSDS	SSTS	SSTS	SSTS	SSBS	SSTS

SSTS, Single-stage top suction. SSDS, Single-stage double suction.
†Large capacity pumps proposed. *Operating pumps. **Constructed (water tested).

The PFBR pump shows considerable reduction in size and weight compared to the French SPX-1 (SUPERPHENIX) pump of similar capacity. It is similar to the Russian BN-600M version, but less complicated because of the use of simple hydraulic layout. It may be noted that only SPX-II pump (a third-generation pump design from France) has better characteristics; however, this design remained on the drawing board.

Revised cavitation criteria

The early fast reactor pumps designed abroad employed a large margin between $NPSH_A$ in the circuit and $NPSH_R$ at 3% head loss by ensuring that NPSHA was larger than the NPSH when bubbles are visible, i.e. NPSHA > $NPSH_{vis}$. Although this resulted in pumps with long impeller life, the size of the pumps remained large resulting in increased capital cost because the pump size directly influenced the size of the main vessel in a pool-type reactor.

On the basis of experience gained over several hundred reactor years, the French revised the NPSH margin for pumps of the SPX-2 (1500 MWe) plant. For these pumps they proposed a cavitation criteria based on $NPSH_{0\%}$ head loss plus a margin to be confirmed, i.e. $NPSH_A > NPSH_{0\%} + X$ (margin). Erosion experiments in sodium conducted on an impeller of a secondary pump showed negligible erosion at $NPSH_{0\%}$ + 10% in a test conducted for 1350 h. However, subsequently for the European fast reactor (EFR) planned by the UK, France and Germany, an NPSH criterion was defined as formation of a 10 mm cavity in 1/5 scale model in water. While it could be inferred that the NPSH at which measurable erosion damage occurs, viz. $NPSH_{er}$, was lesser than $NPSH_{vis}$ by 10%, there was no relation established between $NPSH_A$ and $NPSH_{er}$.

Nevertheless, at IGCAR, a decision was taken to develop hydraulic model to meet $NPSH_{er}$ below NPSHA by certain margin to assure long life of impeller and related parts of pump supplier and the contract. An $NPSH_{3\%}$ corresponding to $NPSH_A/NPSH_{3\%} = 1.5$ was specified in the contract with the pump supplier, and this was much lower than that used in the earlier developed model (1.8/2) impeller.

Development of pump hydraulics

On deciding the above revised cavitation criterion a new hydraulic development contract was worked out with the erstwhile supplier, the scope of which was limited to only model development and testing. No testing of prototype hydraulics was considered, because our earlier experience had shown that the performance of prototype hydraulics could be well predicted from model test results. The tests were aimed at not only establishing a design with high efficiency and stable performance (*H*–*Q*) curve, but also included detailed cavitation performance testing. The cavitation testing comprised of establishing conventional $NPSHR_{3\%}$ versus flow curves, and additional ‘paint erosion’ tests. Identification of cavitation-susceptible regions on impeller suction using ‘paint erosion’ tests was being carried out for the first time. Therefore, a suitable paint had to be developed which did not erode under normal flow (non-cavitating) conditions. Such a paint was not readily available in the local market and there was no response from foreign suppliers. A survey of similar work in this area revealed that BHEL hydromechanics labs had developed similar paints for their work on water turbines. In collaboration with BHEL, several trials were made using the earlier 1/3 model and a ‘special’ paint composition was finalized[16]. Further tests were carried out on the same model in the laboratory of the pump supplier. These included: (a) visual tests to study bubble growth with

Table 5. Performance results with various impeller–diffuser combinations[34]

Impeller and diffuser combination	Performance at duty point flow Q = 508 l/s, H = 60 m, η = 87%, NPSHR$_{3\%}$ = 8.4 m			Remarks
	H (m)	η (%)	NPSHR$_{3\%}$ (m)	
I1B1	62	81	9.8	η less, NPSHR higher, BEP shifted towards right-hand side.
I2B2	63	82	9.5	η less, NPSHR higher, BEP shifted towards right-hand side.
I2B1	64	85	9.9	η less, NPSHR higher.
I3B3	60.5	86.9	7.9	NPSHR achieved is 7.9 m, which is less than the specified value of 8.4 m. η and total head achieved according to requirement.
I4B3	61.5	83	9.5	η less, NPSHR higher.

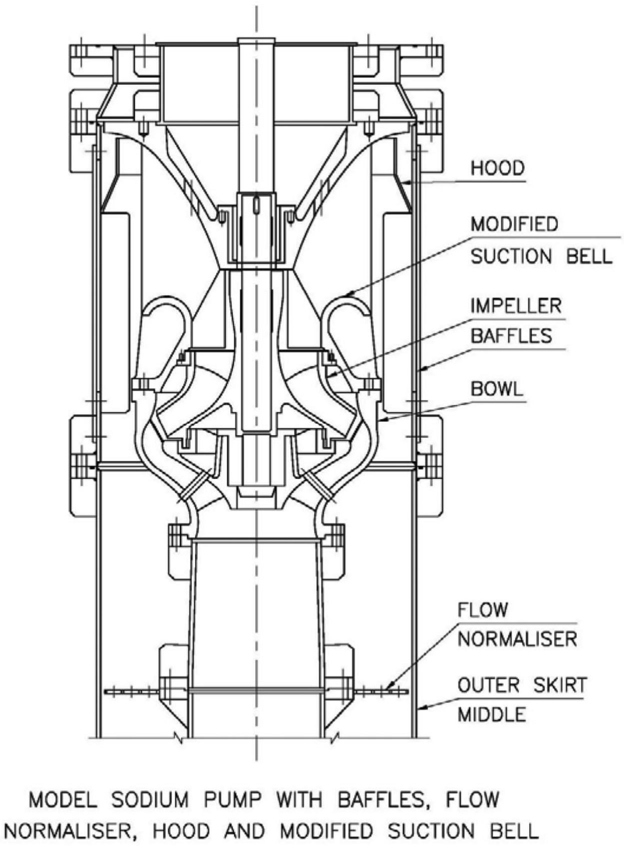

Figure 13. Model sodium pump (modified).

NPSH; (b) paint erosion tests to identify cavitation-susceptible areas in pump suction; (c) tests to study the variation of eroded areas with NPSH, and (d) buffed specimen erosion tests. The buffed specimen erosion tests consisted of long-duration operation of pump under cavitating condition with spot-welded buffed stainless steel specimens on the cavitation-susceptible areas (identified by paint erosion tests) of the impeller, so as to quantify erosion damage.

Hydraulic tests on revised scale model: A 1/2.75 scale vertical model of the pump operating at 1450 rpm was chosen for hydraulic performance validation in water. The tests included determination of performance curves (i.e. H versus Q, P versus Q, efficiency versus Q and NPSHR$_{3\%}$ versus Q) followed by detailed cavitation testing, viz. visual measurement under stroboscopic lighting of cavity growth with NPSH and paint erosion testing. The test loop was designed to be leak-tight and provided with cooling facility to maintain the water temperature below 30°C. Initial tests on different impeller/diffuser combinations were done to select the best combination that yielded a stable H/Q curve with high efficiency and low NPSHR$_{3\%}$. In all, four impellers and three diffuser designs were tried out in five unique combinations and the combination with the least NPSHR$_{3\%}$ (well below the specified value) selected (viz. I3B3 in Table 5).

Testing of selected impeller–diffuser combination: In order to facilitate visual observation of cavity formation at impeller inlet, the impeller shroud, suction passages and intake skirt of the vertical model were made in transparent sections (Figure 13). During the initial tests rope-like submerged vortices were observed at the pump suction. This was attributed to the 90° bend of inlet piping entering the vertical pump model and partly due to the 180° turning of flow at the eye of the impeller. The vortices were eliminated by (a) introducing a flow normalizer at the bottom of the vertical straight portion before the impeller inlet, and (b) fixing eight vertical ribs on the intake skirt for destroying any pre-swirl in the inlet path and improving flow guidance. During the subsequent paint erosion tests, further improvement in the suction casing profile was found necessary. The profile was modified to remove dead zones and ensure smoother entry of liquid into impeller eye. Figure 13 shows the final arrangement used for further testing.

Paint erosion tests were carried out using special paint developed along with BHEL, and the procedure finalized with the 1/3 model pump. The paint was applied on impeller vane passages in three layers with the topmost layer being a protective coating against high-velocity flowing liquid. A grid structure of size 10 × 10 mm was plotted on the blade surface; this provided a reference scale for measuring the eroded area and especially for estimating the cavity length under stroboscopic lighting.

Visual testing for cavitation patch measurement was carried out to measure the growth of vapour cavity with lowering of NPSH. The NPSH was varied and a plot of cavity length versus NPSH was generated. Figure 14 shows variation of both cavity length and eroded paint areas with NPSH.

The pump was then operated at specified NPSH value for 6 h, after which the impeller was dismantled and examined for erosion. The experiments were repeated for design flow and few more flows both above and below design point. Figure 15 shows the impeller after a typical paint erosion test.

The cavity length variation shows an inflexion point, which is similar to the increase in pump cavitation noise with developed cavitation. Although the cavity length continually increases with reduction in NPSH, the paint erosion curve shows a decrease after substantial reduction in NPSH. This is due to the cushioning of bubble collapse by the rapidly increasing bubble cavity volume and is similar to the observed variation in cavitation noise with NPSH.

The measured cavity length in the model pump at NPSH value of 12.8 m (this corresponds to a value of 16 m (which is the plant NPSH) in the prototype) is 34 mm. The corresponding cavity length in the prototype increases by the scale factor and is 93.5. Using empirical relation proposed by Gulich and Pace[17], the life of the prototype impeller operating at plant NPSH was estimated to be 20 years (ref. 18), which is a reasonable value. It is therefore concluded that the selected impeller/diffuser combination will have insignificant erosion when operating under plant NPSH.

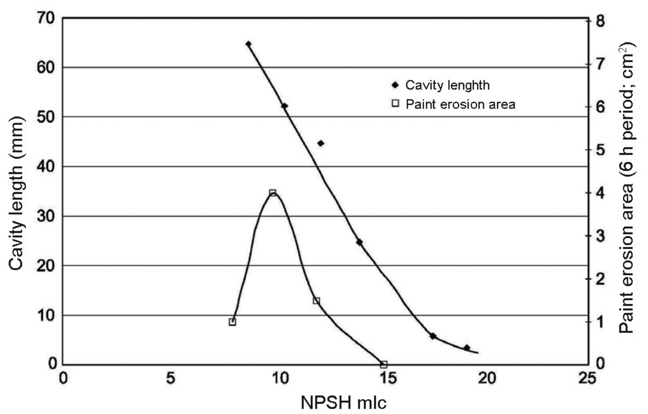

Figure 14. Cavity length and eroded area versus NPSH.

Figure 15. Paint erosion results.

Mechanical components development

The pump is supported on the roof slab at the top and engages with reactor piping at the bottom. This results in the top and bottom of the pump operating at widely different temperatures and the temperature differential varies with reactor power. The axial expansion of the pump was accommodated using a pump–pipe receptacle, while tilting of the pump resulting from the radial thermal expansion was accommodated using a compliant top support consisting of a large commercially available spherical bearing. A commercially available geared spacer tube or 'disc' tube coupling ('flexible' coupling) was chosen between the pump shaft and drive motor shaft to take care of combined parallel and angular misalignments of 6 mm and 0.5° respectively.

Furthermore, it was proposed to preset the pump axis in cold condition such that it remained nearly vertical during nominal operating temperature of 400°C but was slightly inclined at other temperatures, such as during fuel handling or initial commissioning. This also limited the load on the bottom bearing during operation.

The feasibility and robustness of all the above features under the demanding operating conditions had to be experimentally validated before being implemented.

Inclined rotor testing[19,20]

In order to evaluate the performance of the top spherical bearing and the bottom bearing under inclined condition, a near-full-scale rotor test rig was built. The pump impeller was substituted by a disc of equivalent mass and unbalance, and a 125 kW drive motor was used. Three slave pumps were used to feed pressurized water to the bottom hydrostatic bearing corresponding to the pump speed. Figure 16 shows the test set-up. The maximum rotor design speed was 700 rpm (somewhat higher than the pump nominal speed) and rotor unbalance kept at 560 g-cm equivalent to ISO grade 2.5. The weight of the rotor assembly was 100 kN (10 tonnes) and critical speed of the assembly was estimated to be 1050 rpm. The top bearing region was provided with non-contact eddy current sensors to measure shaft eccentricity during operation. Additionally, vibration sensors were provided to monitor vibration at the upper bearing support, spherical plain bearing and bottom pump–pipe receptacle. The journal movement within the bearing sleeve (fixed) showed a loci plot (Figure 17). The journal loci plots obtained at different inclinations of the rotor were similar, indicating minimal effect of inclination on the shaft movement. Vibration readings recorded during continuous operation of 24 h duration and cumulative operation of 200 h duration showed acceptable values. The coefficient of friction at the upper spherical bearing support was determined to be 0.23, typical of sliding friction. A pulling force of 3000 N was required to tilt the rotor assembly weighing 100 kN.

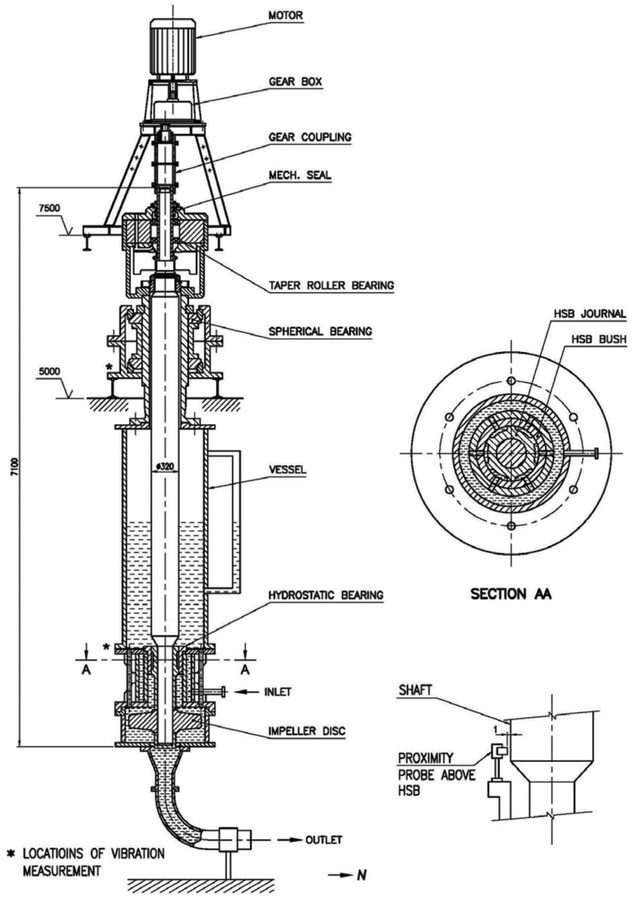

Figure 16. Test section of sodium pump development test rig.

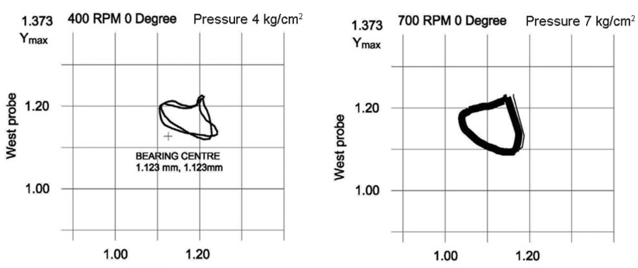

Figure 17. Orbit plots at 400 and 700 rpm.

Testing of pump–pipe receptacle[21]

Another important area that demanded attention was the design of pump–pipe receptacle at the bottom of the pump. This joint is the interface between pump discharge nozzle and the pipe leading to the reactor inlet plenum. The joint is to be designed for easy assembly and removal of the pump from a height of 11 m with a misalignment of 30 mm. However, it has to ensure minimal bypassing of high-pressure sodium into the cold pool, cavitation-free operation, accommodate axial thermal expansion of the rotor assembly and provide adequate guidance to pump discharge nozzle to minimize vibrations. The joint was studied experimentally on a 1 : 1 scale in water and leakage flow was 130 m^3/h at full pump discharge head of 75 m and a joint clearance 0.6 mm. Tests were also performed with the pump–pipe receptacle in inclined position thus simulating pump tilt. The leakage flow in the latter case was observed to be less than that in the aligned condition due to non-uniform clearance affecting the hydraulic diameter (reduced). Prior to this experimental verification, another scale model of the joint (1/2.5 on dia, 1 : 1 on pump height) was built in-house and all mechanical assembly aspects thoroughly verified.

Manufacture and testing of integrated prototype primary sodium pumps

Challenges during manufacturing[22]

The earlier experience of manufacturing the 7.1 m long shaft proved to be useful in tackling the challenges during heat treatment. The shaft of the primary sodium pump (of austenitic stainless steel AISI304LN) is of composite construction and consists of a hollow central portion welded at either end to solid sections. It is ~11.3 m in length with a maximum diameter of 630 mm in the hollow central portion. The shaft assembly is supported at the bottom by hydrostatic radial bearing and at the top by tilting pad type thrust bearing and sleeve type radial bearing. The shaft was stress-relieved in an electrically heated furnace in controlled reducing atmosphere of argon with 10% hydrogen. The slight reducing furnace atmosphere ensures bright surface after heat treatment. Balancing after heat treatment was done to grade G 0.4 (ISO 1940) at a speed of 300 rpm. The bush and journal of the hydrostatic bearing were hard faced with Colmonoy 5 to provide adequate resistance against rubbing during start-up and coasting down of the pump. The deposit thickness was ~1 mm and the measure hardness of the surfaces was 45–47 HRC. Detailed vibration analysis of the rotor assembly was done to confirm that the natural frequency of the rotor deviates from the operating speed by more than 20%.

The impeller was machined from imported casting that satisfied the requirements of ASTM quality level-2 (no cracks, hot tears, inserts and mottling) with surface finish better than 6.3 μm.

Testing of primary pumps in water[23]

Performance testing of all the three pumps was done in water. The following tests were carried out:

- Measurement of flow versus head at nominal and two other speeds.
- Measurement of $NPSHR_{3\%}$ at various flow rates.
- Operation of the pump with shaft in inclined condition.
- Endurance testing of 50 h duration at nominal operating speed.

Testing of the pump was done in an open sump at the manufacturer's test facility. During the initial tests, it was observed that flow pattern at the pump suction was non-uniform and pump performance was affected. The problem was investigated and the sump modified in conformance with ANSI HI 9.8 (1998), height of breast wall increased and additional webs introduced in the pump suction to break vortices. Water quality in the sump was controlled and filters provided at sump entry. A cooling system was also provided to limit the temperature in the pool to 40°C in view of the high (4.2 MW) power input.

The NPSHR 3% tests were done by varying the water level in the pump suction and recording the break-away point from the non-cavitating *H–Q* curve. Accelerometers were used to record vibrations at the top bearing location. The measured values of 2–3 mm/s were within the satisfactory limits specified in ISO 10816 for large pumps on soft foundation.

Performance tests were run at different speeds, including the lowest operating speed of 98 rpm envisaged during fuel handling stage or during any situation where pony motor is used to run the pump. Figure 18 shows the performance curves.

It was observed that the head developed by the pump (84 m) at the rated flow was higher than that predicted from model tests (75 m). This may be attributed to the modification done in the suction casing to reduce the swirl at the inlet resulting from the 180° turn of the flow. The modification consisted of introduction of additional ribs in the suction casing and was warranted to overcome problems encountered during the initial performance tests.

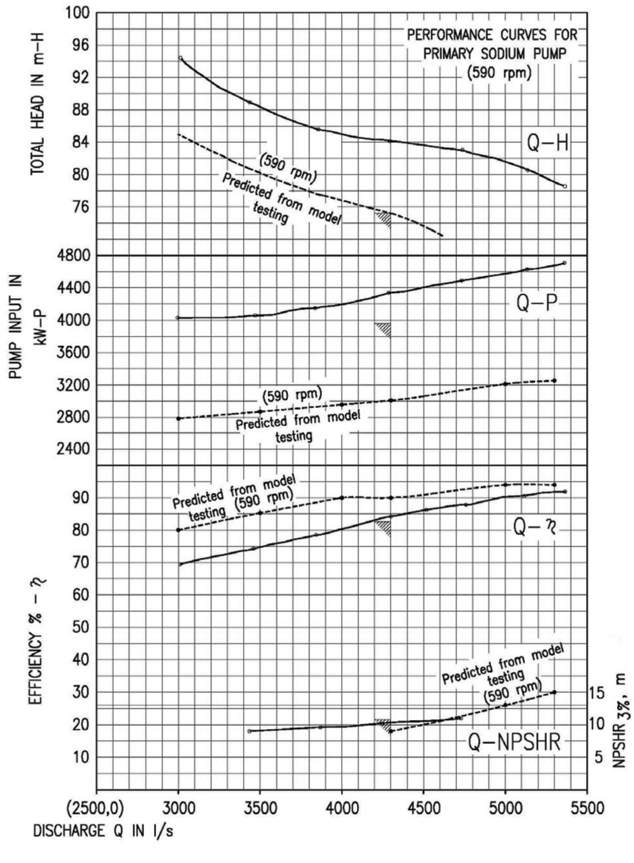

Figure 18. Performance characteristics of prototype primary sodium pump.

Inclined rotor testing[24]

The primary pump was installed with inclination of 0.14° in the reactor at room temperature. It was operated at varying degrees of inclination at different power levels

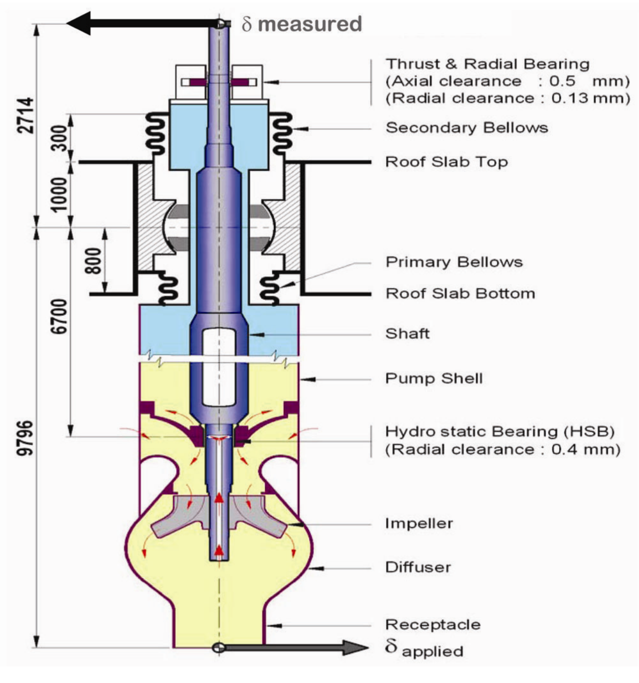

Figure 19. Arrangement for inclined rotor testing of prototype primary sodium pump.

Figure 20. Prototype primary sodium pump being assembled in test bed.

and was maintained vertical only at the normal operating temperature of 670 K (i.e. at the rated power). The pump was therefore tested in inclined condition. Figure 19 shows a schematic of the test arrangement. The tests provided valuable inputs in improving the performance of the gear coupling as well as design of the secondary bellows. The pump was operated at nominal speed for 4 h and performance was found to be satisfactory.

A 50 h endurance test was carried out at the nominal operating speed to verify continuous hydraulic and mechanical performance of the pump, including that of bearings and seals. The performance was satisfactory.

After completing all tests, the pump was stripped down and inspected for any rubbing marks/damage on bearings and seal surfaces, and found to be satisfactory.

Figure 20 is a photograph of the pump under assembly at the test bed.

Summary

In this article we have covered all the important aspects and activities in the development of liquid metal (sodium) pump for a large fast reactor power plant (PFBR). Though the total span of the development activity (over a period of 15 years) may appear rather long on the face of it, the extended span was primarily due to change in the pump ratings and arrangement midway through the work (resulting from revision in the reactor design from the erstwhile four loops concept to two loops concept) necessitating additional testing. It may be appreciated that the detailed cavitation study carried out on the pump model is first of its kind in the country.

Particularly noteworthy in this development work is the successful collaboration between a government research centre and the industry in an area such as pump development, where information sharing is not the norm. It is our conviction that the experience gained in this exercise of joint pump development will be useful whenever similar tasks are attempted in the future.

Notes

1. The heat transport system in a fast reactor consists of a radioactive primary circuit, an intermediate secondary sodium circuit and a tertiary steam/water circuit. The heat generated in the nuclear core is transferred by liquid sodium circulating in the primary circuit to sodium circulating in the secondary circuit through intermediate heat exchanges. The heat thus transferred to sodium in the secondary circuit is used to generate steam in the tertiary steam/water circuit using once through steam generators.
2. Bowl efficiency is the efficiency calculated on the basis of the measured flow and head at the outlet of the diffuser (bowl). In the reactor pump, the actual flow will be less than that at the exit of the diffuser because (i) a small fraction of the flow is used to provide bearing cation (in the hydrostatic bearing), and (ii) another small fraction leaks into the pool through the labyrinth at the pump–pipe connection.
3. Suction specific speed is a measure of the suction capacity of a pump (i.e. the ability to operate with reduced NPSH). It is given by the formula $N_{ss} = N*Q^{0.5}/H^{0.75}$, where N is the pump speed (rpm), Q the pump flow (m^3/s) and H is the pump head (m).) of 218 based on required 3% NPSH ($NPSHR_{3\%}$).
4. Alternate design options such as multi-stage or double-suction impeller were not considered as these tended to further complicate the hydraulics. On the other hand, an inducer pump, normally used in limited NPSH environment, was not contemplated because the inducer acts as a sacrificial impeller and this can result in contamination of reactor sodium by debris.

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