GAS FLOW TURBINE POWER GENERATOR FOR AUTONOMOUS SUBSEA PRODUCTION SYSTEMS

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ABSTRACT

Tecnomare is developing a subsea in situ energy generation unit, as part of the Deepwater Autonomous Multiwell Production System (DAMPS) project. A simple, compact turbogenerator has been identified. It exploits the head drop of the gas flow across a turbine under practically incompressible flow conditions, to generate electric energy. It will be applied downstream the manifold in gas production or oil production systems integrated with subsea separation and test units. The operational requirements can be fulfilled by existing commercial process pumps, of multistage radial flow type, slightly modified and integrated with purpose built electrical components. The unit is totally closed and resistant to fluid's pressure; therefore all moving parts are protected from the sea environment, and any leakage towards the environment itself is prevented.

A full scale prototype will first be tested to confirm performances. This will be subsequently field proven on a surface production facility, to ascertain the required reliability and life, before its final application on a subsea production systems.

The paper describes the selection criteria, the system architecture and the results of the preliminary analysis of the pump-turbine performances.

INTRODUCTION

Tecnomare is undertaking a research project for the development of a Deepwater Autonomous Multiwell Production System (DAMPS), with the financial contribution of the EEC and the participation of AGIP SpA.

Previous activities of Tecnomare in this field were to develop acoustic systems for the control of subsea Xmas trees [1,2]. This technology and system proved effective and promising for satellite wells and field extensions.

The energy supply for the first generation single well system was based on primary Lithium batteries, which is an acceptable solution for near term demonstration, but would not be optimum for operational multiwell systems. Unless a very large capacity is provided the system would require costly intervention for the replacement of the discharged batteries.

The DAMPS project, among other issues, has as its main goal the development and the experimental demonstration to prototype level of an in situ, autonomous and continuous power generation system. A conceptual design phase considered and evaluated various alternative systems, among which the following were shortlisted:

- Closed Organic Rankine Cycle systems (ORC) utilizing as a heat source the fluid flowing through the production system, either gas or oil.
- Thermoelectric systems, utilizing the same primary energy as the above, but with static components;
- Seawater batteries, utilizing a controlled corrosion reaction through seawater, between two electrodes [3]
- Turbogenerator, utilizing the hydraulic or kinetic head of the high pressure fluid.

The preliminary analysis, taking into account also the state of the art of the respective technologies, concluded that sea water batteries are most competitive for oil production whereas turbogenerators are most promising if working with a gaseous fluid, namely for gas production or oil production systems with medium to high GOR, for which a subsea well testing system is foreseen[1]. The advantages and disadvantages of the above candidate systems are summarized in Table 1.

As a conclusion of this phase, given the prospects on future gas fields developments it was decided to proceed with the development of the Gas Turbogenerator System described in this paper.

SYSTEM DESCRIPTION AND REQUIREMENTS

The architecture of the DAMPS and the autonomous power generator system is shown in Figs. 1, 2 and 3. The power system is composed of the following elements:

A) Separation unit to eliminate solid/liquid particles, in parallel to a choked by-pass which establishes a pre-defined differential head across the turbine;
B) Self contained turbo-generator subsystem;
C) AC/DC converter and control electronics, inside a pressure vessel, filled with inert gas at atmospheric pressure;
D) Rechargeable lead acid battery stack, as energy buffer against peak loads;
E) Electrical and process lines and mechanical connectors as the interface with the other DAMPS' subsystems.

All these units will be installed in a suitable skid as a one piece module to be lowered/retrieved onto/from the production template by means of an installation and reentry system. This will operate

[1] This technology is also being developed in another EEC funded project by Tecnomare in collaboration with AGIP, Snamprogetti and CALTEC.

the connectors remotely.
The multistage turbine is driven by the flow of the natural gas and, in turn, drives the electric generator, directly coupled to its shaft.
The power is generated by a 3 phase alternator and is rectified to feed the DC electric loads of the underwater completion.

Energy demand

The operation of an autonomous subsea production system demands for various and changing levels of power, following the functions that are activated. These may be summarized as follows:

A) Continuous loads

- System monitoring functions, i.e. well instrumentation, system's status;
- Communications receiver controller (listening system);
- Fail safe bleed-off system power supply;

B) Peak loads

- Hydraulic system recharging for tree valves actuation, during functional check-out [1,2];
- Acoustic communication transmission;

A typical power demand diagram is represented in Fig. 4. The base power demand corresponding to said diagram is 65 W for a five well system.
The electric generator shall satisfy the following two functions:

a) charging and mantaining charged, by means of the battery charger, the battery stack after peak load discharging;
b) delivering a continuous power of 65 W to the DC loads of the underwater completion, electrically connected to the battery.

The temporary, peak load requires a battery of 125 Ah, at 24 V; the relevant 12 hour recharge power is thus 300 W, with a full charge voltage 1.2 times the nominal. Considering losses in the rectifier and battery recharge system, a total continuous requirement of 410 W results as the output from the electric generator, or 550 VA with a power factor of 0.75. Assuming a generator efficiency of about 90 %, the power requirement at the turbine shaft is therefore 450 W.

Operational requirements

Gas pressure, temperature and capacity will change during field life. Table 2 shows the minimum, nominal and maximum values for said parameters. The power generation unit shall also comply with the pressure rating of the whole system, in particular of the sealine. A 210 bar (3000 psi) rating is considered for the first demonstration prototype; for subsequent commercial systems a 350 bar (5000 psi) rating shall be possible on necessity.

To avoid excessive changes of the gas flow rate available to the turbine during the life, the installation of the turbogenerator is in or downstream the manifold; the turbogenerator could alternatively be installed upstream the manifold, but in this case power supply would not be satisfied, should the well delivering gas to the turbine be shut-off for production or maintenance reasons. Multiple machines could be foreseen, but complexity would increase.

Long exposure times to sea water ask to avoid wet moving parts and to use materials resisting corrosion from the outside; in the innerside some parts are also exposed to corrosive sour gas, threatening the material with Sulfide Stress Cracking (SSC) and pitting [5].

Finally a minimum Mean Time Between Maintenance and Repair (MTBM and MTBR) of 3 years or 25000 hours is a mandatory goal to achieve cost effectiveness in comparison with other candidate systems. Indeed intervention costs would be more costly than the power generation module itself.

From the above, reliability and maintainability are clearly key success factors, of much higher importance than other performance characteristics of the machine and of the whole module. These issues suggested maximum utilization of state-of-the art technology and components to be integrated and applied to this system, although properly adapted and improved with minimum modifications.

Turbine

The turbogenerator is made of a commercial process pump, applied also as a water injection unit in the oil industry, interfaced with a purpose built sealed electric generator (Fig. 5), cooled by natural convection.

The pump casing is axially split, designed to withstand the rated pressure (210 bar) and hydraulic tests at 350 bar. Its impeller is a four stage radial flow vane with 250 mm outer diameter, axially balanced. To limit the power output within the required value, it can be destaged by replacing two of the exceeding vanes with suitable axial spacers. This modularity allows high flexibility with respect to operational and output conditions specification by using the same design and components. The pump-turbine is designed and certified following API 610 standards [4].

The maximum working speed of the commercial model is 5000 rpm. However it will be used at 1500 rpm in operation as a gas flow turbine, which allows a higher reliability and life of the roller bearings which are the sole components subject to wear. They are housed within boxes in the pump casing partially filled with lubricating oil, pressure balanced by communication with the gas flowing through the turbine. Labirinth seals are therefore sufficient to prevent losses of the lubricant from these boxes along the shaft. Coupling between the turbine and the generator is direct, as it will be explained in the following, with no rotating shaft exiting from the pressure resistant casing.

This design avoids high pressure rotary mechanical seals, subject to wear and generating potential leakages and high power losses.

Electric generator and stationary equipment

The electric machine is an AC generator made of a rare earth permanent magnet rotor and a stator

enclosed in a pressure resistant housing filled with dielectric oil.
The rotor, directly mounted onto the pump-turbine shaft, rotates inside a thin stainless steel cup placed in the gap zone of the electric machine and flanged between the pump casing and the generator housing. This cup allows maintaining physical separation between the two different environments: the highly corrosive natural gas (which can intrude into the rotor side) and the inner part of the electric machine (the stator side) filled with silicon oil; this is pressurized by the natural gas by means of a compensation system. This allows satisfying safety requirements, preventing that stator's materials and electric windings are in contact with gas and avoiding the possibility that an explosive gas mixture enters the electric generator; this might have otherwise occurred as a consequence of air intrusion when the system is retrieved at surface for maintenance.
The compensation system allows also to require resistance to pressure load only to the external housings of the machine.
The generator's housing is designed to exchange heat losses by natural convection either with water or air; the latter case is foreseen for test purposes or eventual application of the unit on a platform or land facility. No forced circulation of the coolant is needed, thus related fault sources are avoided.
Due to the variability of energy demand, a rechargeable battery stack is interposed as a storage buffer between the AC generator and the DC template loads. The battery stack is charged and maintained charged by a regulated AC/DC converter with a "crossbar" output characteristic. This is composed of a power and a signal section.
The power section consists of a rectifier and a smoothing filter for the conversion from alternate to direct current and of a regulated DC/DC conversion power transistor of the switching type.
The signal section consists of circuits to drive the power transistors and of control loop for the regulation of the output voltage and current. The battery charger will work in two modes, either at constant current or at constant voltage, the latter being used when full charge is reached; the output characteristic of the battery charger will be of the constant voltage type for trickle charge conditions of the battery.

Materials

Particular attention has been paid to specification of the materials used for the realization of the system components. Indeed both the inner and the outer side of the pump-turbine are subject to aggressive corrosion, respectively by the sour gas environment and the sea, or the offshore atmosphere, should the system be at surface for tests or maintenance.
Gas conditions are such as to create potential Sulfide Stress Cracking problems. All parts subject to high stress and in contact with natural gas, like the shaft and the housings, are thus made of duplex (austenitic/ferritic) stainless steel. Hardness requirements, no mechanical seals being required, are indeed compatible with the application of this material, following NACE standard MR-01-75 [5].
Other parts of the machine are made of AISI 316 or 316/L stainless steel.

EXPECTED PERFORMANCES

The turbine to exploit the available gas flow shall have a *specific speed* complying with the boundary plant's conditions, namely the flow rate available and the power required, according to the following formulas:

$$H_t = \frac{P_t}{\eta_t \, \rho \, g \, Q_t}$$

$$n_{st} = \frac{n_t \sqrt{Q_t}}{H_t^{\frac{3}{4}}}$$

where:

n_{st}	turbine specific speed referred to flow capacity and head (S.I. units)		
H_t	total head drop across the turbine[1]		m
n_t	revolution speed	1500	rpm
Q_t	volumetric flow capacity	$21\ 10^{-3}$	m^3/s
P_t	turbine shaft power	450	W
η_t	turbine efficiency	60	%
ρ	operating fluid density (gas)	56	kg/m^3
g	gravity acceleration	9.81	m/s^2

The above formulas would only apply for incompressible flow. Indeed in the specified conditions gas pressure and density do not decrease appreciably from the inlet to the outlet of the turbine vanes and incompressible fluid-dynamics can be applied with good approximation as will be confirmed in the following.
The volumetric capacity is here referred to the nominal operating conditions, as function of pressure and temperature of the gas. A nominal speed of 1500 rpm is choosen according to the reliability and life criteria cited above, with no requirement for regularity since alternate current generated is rectified.
The head and the specific speed are first estimated by assuming a tentative value for the turbine efficiency (60%). The resulting first approximation values of the above quantities are:

$$H_t = 65 \text{ m}$$

$$n_{st} = 9.5 \text{ (S.I. units)}$$

The n_{st} value corresponds to a relatively slow turbine, say with pure radial flow, single or multistage impeller. The pressure drop across the turbine under the above head is only 0.36 bar, or 0.5 % of the inlet gas pressure. Evaluation of the isentropic expansion work and enthalpy change of the gas leads

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[1] Subscript "t" denotes parameters referring to the *turbine* working condition; in the following "p" denotes *pump* conditions.

to a temperature decrease of less than 1 °C. Both these checks confirm the validity of the incompressibility approach. Therefore the selection of a pump to work as a turbine can be undertaken with reference to similarity relationships and diagrams used for pumps design and selection.
The problem posed by this application consists in answering the question: "Which is the pump type and model and what are its nominal operating data, which, under the specified fluid, flow capacity and shaft speed, guarantee the expected power output?". Among a number of applicable approaches, one in particular has proven relatively simple and effective, which makes use of available charts (see an example in Fig. 7) describing the complete pump characteristics; this is the functional relationship among the four main variables (speed, flow rate, head, torque). Any combination of sign and value of two of them determines univocally the two others [6]. The diagram reports these relationships in non dimensional form, referring all parameters to the respective values at the best efficiency point (b.e.p.) of the machine as a *pump*. Although relatively few such diagrams are available, reference was made to a pump model having a specific speed as close as possible to the value calculated above. Once suitable pump data and commercial models were identified, a more focused analysis and check was performed by contacting qualified manufacturers.
The process utilized in this work can be summarized as follows:

a) the C quadrant part of the diagram of fig. 7 has been translated into the equivalent one shown in Fig. 8 which reports constant power curves instead of constant head curves (as percentage of the value at the *pump's* b.e.p.). The variables and diagram represent *turbine* conditions under reverse flow and speed; they are indicated by the subscript "%" to indicate their value, relative to those at the b.e.p. of the machine as a *pump*;

b) the ceiling speed $n_F\%$ as function of flow rate has then been superimposed in Fig. 8, together with lines of constant ratio between ceiling and turbine speed (n_F/n). This is also an important parameter to be taken into account in pump's selection, to avoid possible damage to the machine, should the load torque become null during operation;

c) criteria for optimum choice have been stated; these are:

   I) the speed of the turbogenerator should vary smoothly at constant power demand, as function of the flow rate; this need is to comply with possible operating flow changes during system life, avoiding large torque variations which would result in demanding design of the electric generator;
   The points complying with this requirement are in the domain defined by

$$T\% \geq 60\%$$
$$n\% \leq 70\%$$

   II) the ratio between turbine's ceiling and operating speeds should be minimum; in particular should be minimum $n_F\%$, referred to the b.e.p. *pump* speed for which the machine and shaft are designed.

The above requirements (I and II) are satisfied by the points A,A',B and C shown both in Fig. 7 and 8, which can be selected for a final comparison.
The parameters of these points, obtained from figures 7 or 8, are reported in Table 3. The corresponding b.e.p. pump data are summarized in Table 4. It must be reminded that the pump data are referred to water pumping, while the desired turbine performances are obviously referred to gas flow.
The result of this preliminary analysis allowed contacting manufacturers, often not having available data on turbine work conditions of their pumps, with a finalized specification to qualify their products for the intended application.
Fig. 9 shows the performance diagram of one commercial process pump which has its b.e.p. very close to case A in the above tables and figures. The upper head curve in fig. 9 refers to the production model with a four stage impeller. The head-capacity diagram calculated from the above for a destaged (two stages) impeller is also plotted (dotted line). It can be seen that the real destaged pump has optimum efficiency between the following two conditions:

$$Q_p = 18 \text{ lt/s} \quad H_p = 55 \text{ m} \quad \eta_p = 69\ \%$$
$$Q_p = 20 \text{ lt/s} \quad H_p = 53 \text{ m} \quad \eta_p = 68\ \%$$
$$(@\ n_p = 1750 \text{ rpm})$$

These data are very similar to the desired values summarized in table 4, obtained by the analysis described.
The pump manufacturer produced also a performance diagram of the same four stage machine working as a water *turbine*, under reverse flow and revolution, at the specified value of 1500 rpm (Fig. 10). It can be seen that with flow capacity of 21 lt/s (75 m^3/h) the four stage turbine power is 16 kW with water flow, corresponding to 900 W with natural gas (see gas power scale). The corresponding power with a two stage impeller is thus 450 W as predicted by the preliminary analysis.
In conclusion the commercial pump identified is capable to produce twice the required power at the specified revolution speed and gas flow rate. This over-performance allows compensation of possible unpredicted features, such as excess of DAMPS power demand during operation or excess losses of the electric subsystem. Otherwise at the end of the trials there will be the possibility to decide about turbine destaging, should this be needed to avoid dissipators of the excess power.

PROTOTYPE DEVELOPMENT AND TEST PLANS

The whole in situ power generation system described above is developed at the basic design level, in a modular configuration suitable for integration in the D.A.M.P.S. system for subsea operation up to 1000 m water depth.
The detailed design is being developed for those units which are foreseen to undergo construction and performance tests, i.e. for the turbogenerator unit, the rectifier and the control electronics; these will

be tested with a standard battery stack.
After construction the prototype will be laboratory tested with a suitable experimental test rig to confirm the expected performances with compressed air flow at the specified density. This phase is expected to end by July 1992.
In a possible second phase of this project the same prototype will be installed on a surface gas production plant, either on- or off-shore, for endurance and reliability tests. Therefore the design of the generator is being developed, considering not only the requirements for the D.A.M.P.S. application but also for installation in hazardous areas, downstream surface manifold and separation units. These requirements are indeed not much more demanding to the design than the subsea application itself and can be satisfied at almost the same cost.

CONCLUSIONS

The in situ subsea energy generation system described in the above is fairly simple and compact. It is also expected to be reliable and cheap, thanks to the solutions identified. The work performed has demonstrated that the necessary requirements can be fulfilled by existing commercial process pumps, of multistage radial flow type, integrated, with purpose built electrical components. The turbo-generator unit is totally closed having a pressure resistant casing; all moving parts are thus protected from the sea environment and any polluting leakage to the environment itself is prevented.
A development plan is in course, which foresees the performance verification of a full scale prototype by laboratory tests on compressed air flow. This will be subsequently field proven on a surface gas production facility, to ascertain the required reliability, before its final application in a subsea production systems.

AKNOWLEDGEMENTS

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CRITERION	ORC	TEG	SWB	GTG
Power range achievable	**	*	*	***
Compactness	**	*	*	***
Applicability to oil fields	***	***	***	**
Applicability to gas fields	*	*	***	***
Insensibility to fluid flow/temperature	*	*	***	**
Prospected reliability/availability	**	***	***	**
Sensib. to susp. solids & paraffins	**	*	***	**
Development costs	*	*	***	***
Development time	**	*	***	**
Prospected on field cost	*	*	**	***
OVERALL EVALUATION			BEST for oil	BEST for gas or high GOR oil

Tab. 1 - Comparison of candidate power generation systems
(*** best, * worst)

	UNITS	MIN	NOM	MAX
WATER DEPTH	m	20	1000	1000
GAS FLOW RATE	Nm^3/h		6000	
	kg/s		1.2	
INLET GAS TEMPERATURE	°C	-10	0	40
INLET GAS PRESSURE	bar	80	80	200
INLET GAS DENSITY	kg/m^3	49	56	146
GAS VOLUM. FLOW RATE (INLET)	lt/s	8	21	24
	m^3/h	29	75	86

Table 2 - Power generation system operational data

CASE POINT			A	B	C	A'
Relative flow capacity	Q%	%	105	97	92	110
Relative torque	T%	%	60	60	60	75
Relative speed	n%	%	85	55	40	70
Relative head	H%	%	92	60	48	88
Relative power	P%	%	51	33	24	53

Table 3 - Non dimensional turbine working parameters corresponding to the reverse flow and revolution conditions of the points indicated in Figs. 7 and 8 (as % of pump's b.e.p.).

CASE POINT			A	B	C	A'
Revolution speed	n_p	rpm	1750	2700	3750	2100
Power (water flow)	P_p	kW	19.3	29.8	41.0	18.8
Flow capacity	Q_p	lt/s	20	22	23	19
Head	H_p	m	53	82	113	58
Specific speed	n_{sp}	(SI units)	13	15	16	14
Efficiency	η_p	%	60	66	69	64

Table 4 - Nominal (b.e.p.) *pump* data, under water flow, corresponding to the points indicated in Figs. 7 and 8 and table 3.

CASE POINT			A	B	C	A'
Revolution speed	n_t	rpm		1500	all cases (tab. 2)	
Power (gas flow)	P_t	W		450	"	
Flow capacity	Q_t	lt/s		21	"	
Head	H_t	m	49	49	54	51
Specific speed	n_{st}	(SI units)	12	12	11	12
Efficiency	η_t	%	61	60	56	59

Table 5 - Turbine working conditions data, under gas flow, corresponding to the points indicated in Figs. 7 and 8 and table 3.

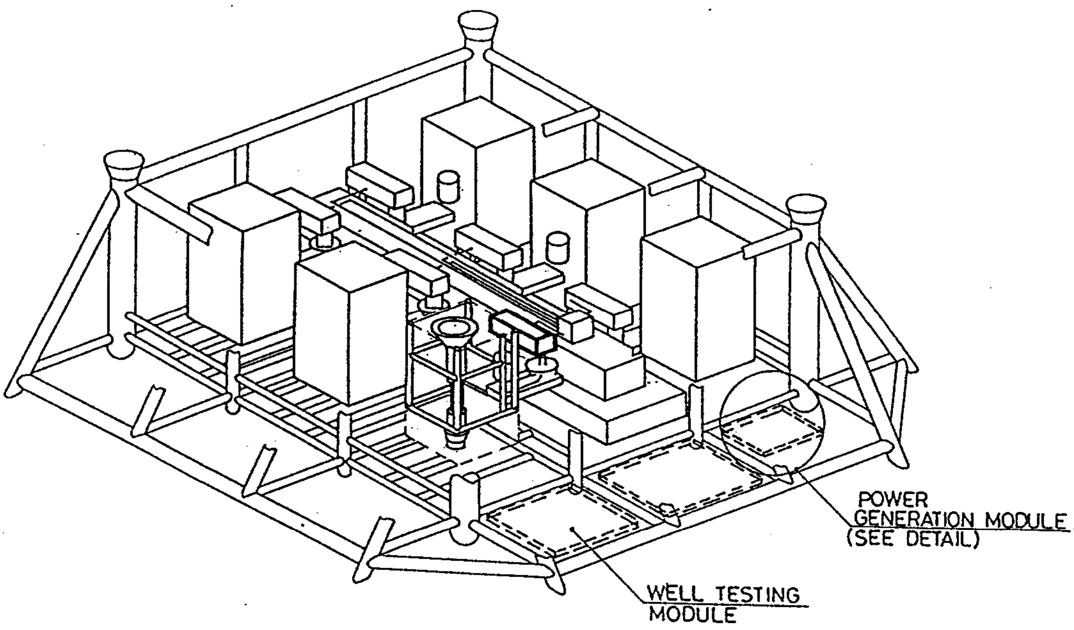


Fig. 1 - DAMPS - General Configuration

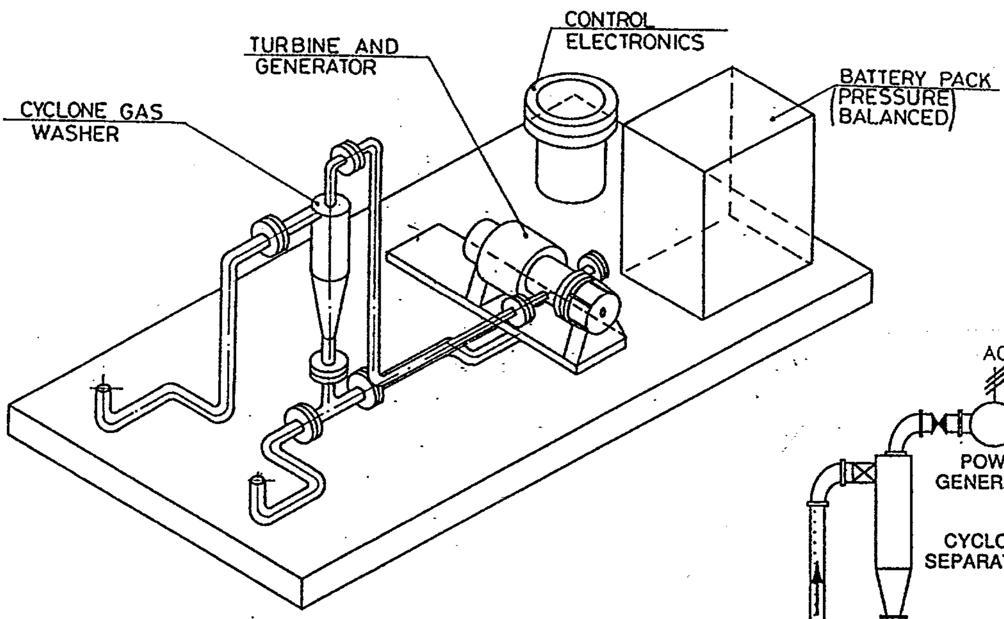


Fig. 2 - DAMPS - Power generation module layout

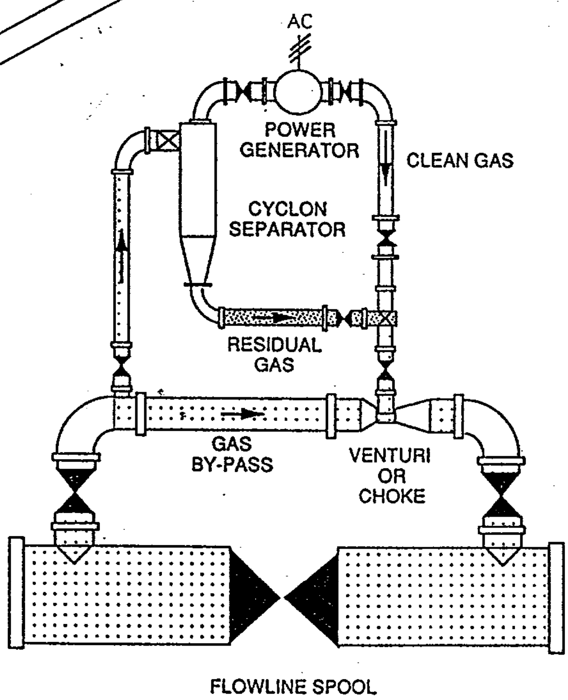


Fig. 3 - DAMPS - Power generator module simplified process flow diagram

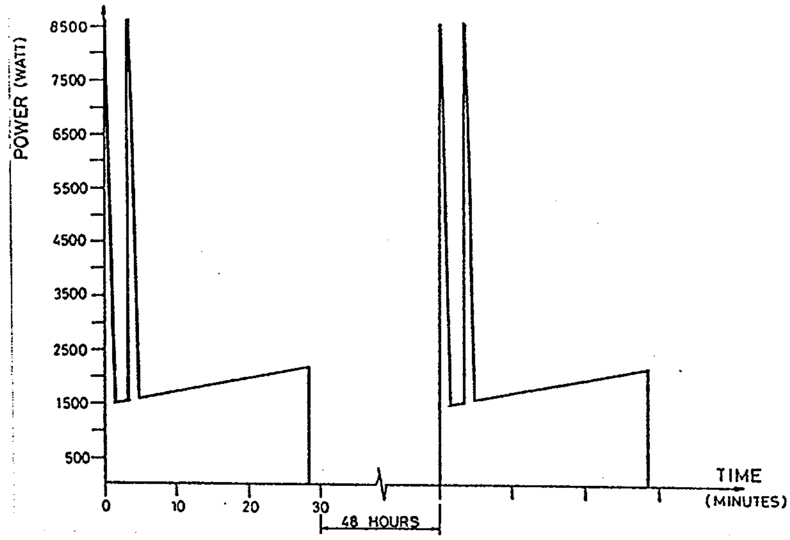


Fig. 4 - DAMPS - Typical power demand diagram

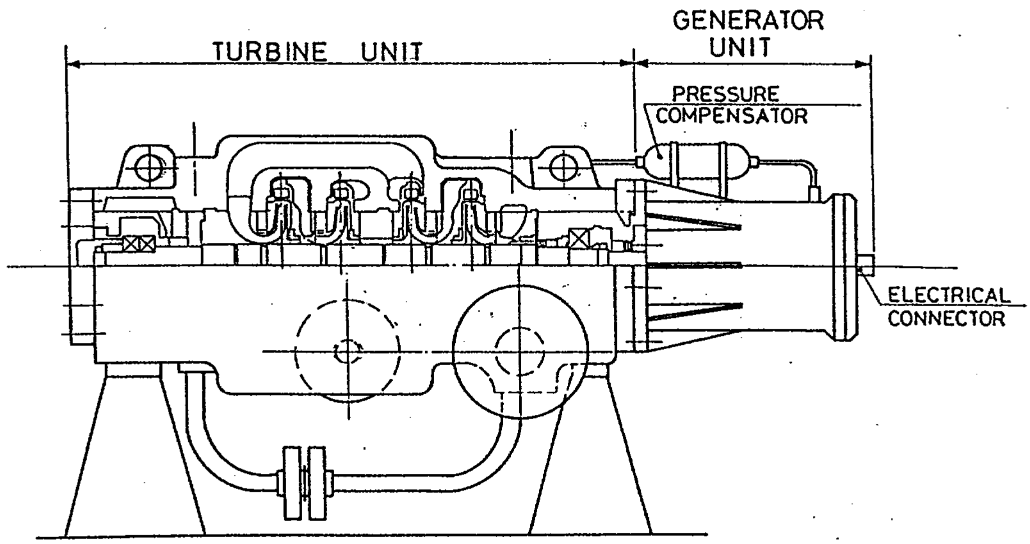
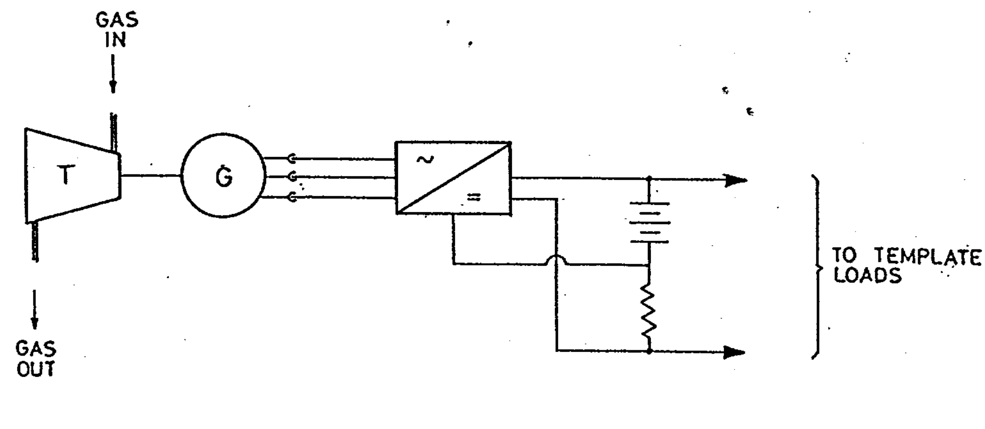


Fig. 5 - Turbogenerator assembly



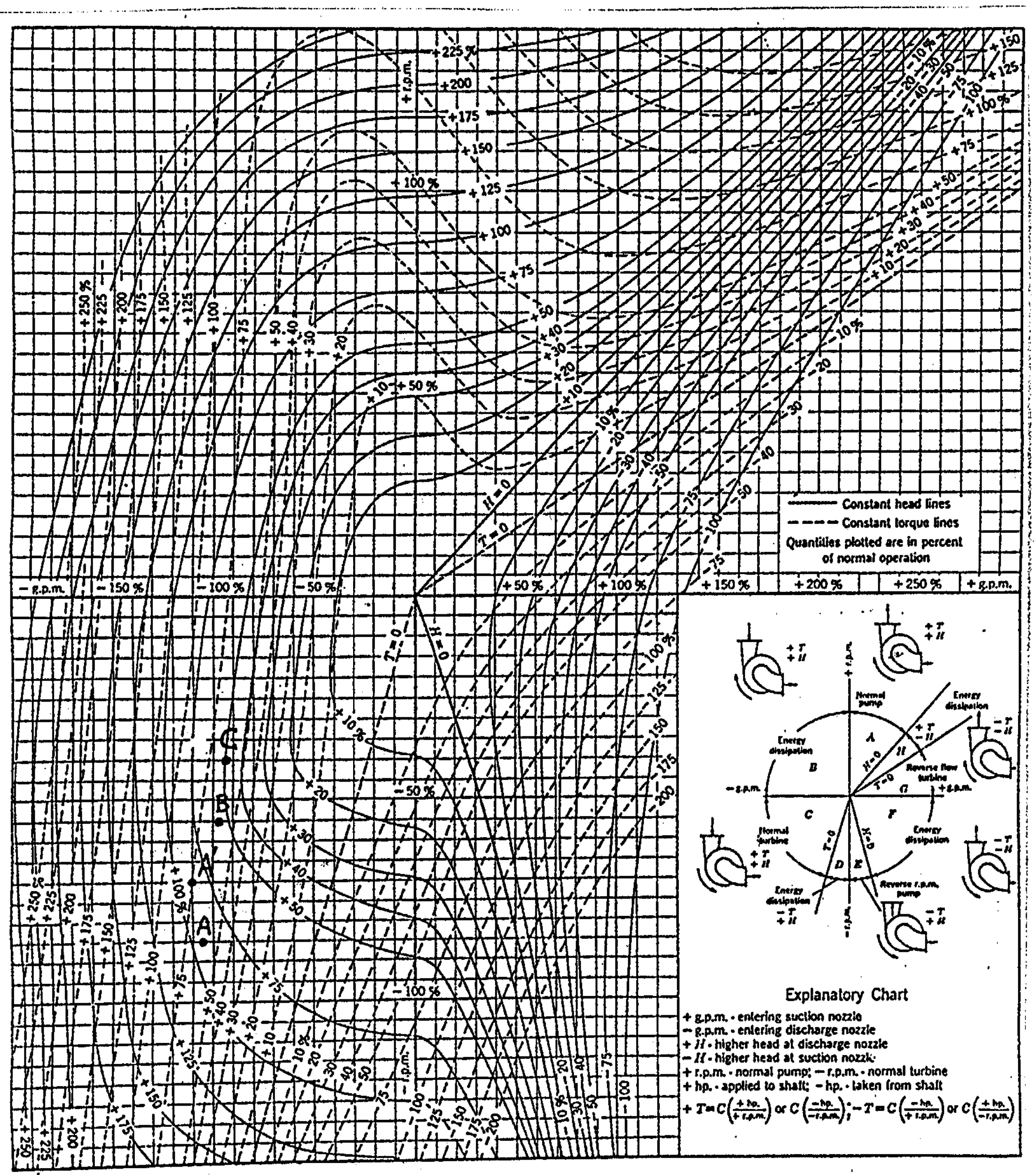


Fig. 7 - Complete pump characteristic chart 6 .
Q% in the x axis, n% in the y axis

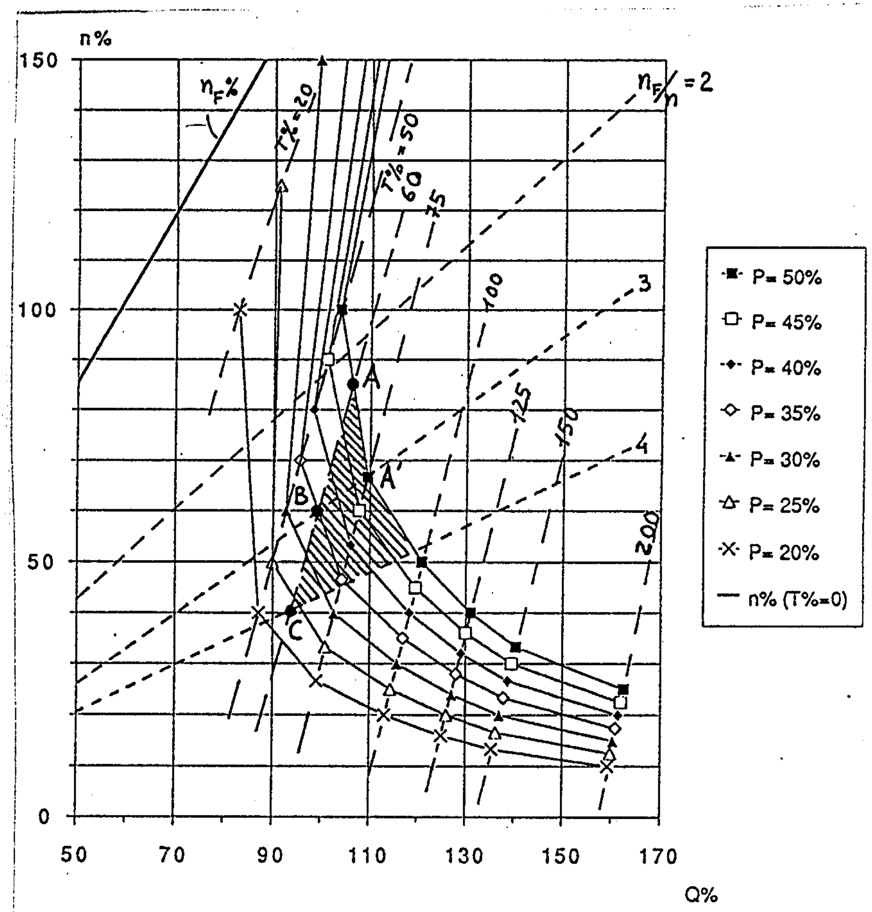


Fig. 8 - Turbine performance chart, derived from Fig. 7

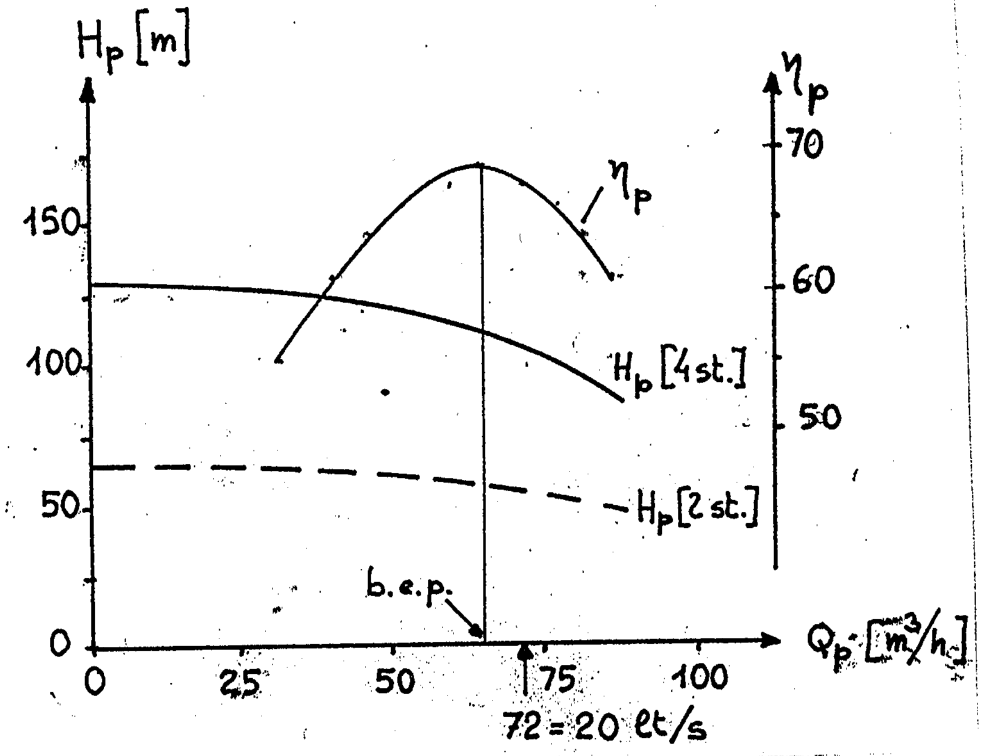


Fig. 9 - Performance diagrams of one commercial process pump (4 stage impeller) and its destaged version (2 stage impeller)

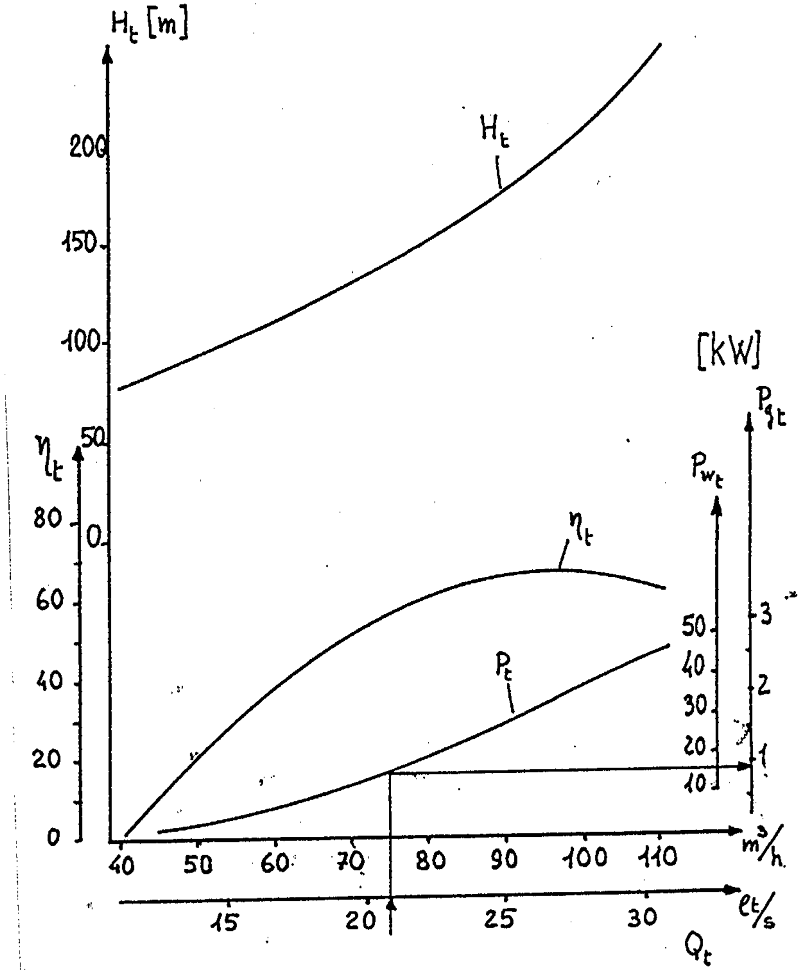


Fig. 10 - Performance diagram of the pump of Fig. 9 as a turbine. Pw and Pg denote power output kW with water and gas respectively