THERMAL ENERGY STORAGE
USING PRESTRESSED CAST IRON VESSELS (PCIV)

FINAL REPORT

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ABSTRACT

The wide-spread application of thermal energy and high-pressure air storage to electric power generation has so far been hampered by the lack of large high-pressure storage vessels of reasonable cost. Welded steel vessels are too expensive for this purpose. However, the Prestressed Cast Iron Vessel (PCIV) - developed as a nuclear reactor pressure vessel by Siempelkamp Giesserei KG of Krefeld, FRG - has the potential of complying with these requirements.

Applications of the PCIV include: high-pressure air storage for the quick start-up of open cycle gas turbines; pressurized high-temperature sensible heat storage by means of solids with a gaseous heat transfer medium for closed cycle gas turbines of future solar power stations; and pressurized hot water storage for nuclear, solar, or coal-fired steam power plants, employing either separate peaking turbines or overloadable main turbine sets.

A reference PCIV of 8000 m^3 , 275°C , with hot going walls and cold going tendons was developed, designed, and stress-analysed. A parametric study showed that pressures between 4 and 8 MPa and L/D ratios larger than 4 should be optimal. Cost of the reference vessel is about \$ 10,000,000 or 33 to 50 \$/kWh electric energy stored. Cost of peak power will be from 30 to 100 mills/kWh, depending on many parameters.

1. INTRODUCTION

Within the energy issue - that has lately become so crucial to mankind - the methods and possibilities of energy storage as a means of matching supply and demand have attracted more and more attention during the last two or three years /1...15/. In particular, the great role that thermal energy storage (TES) is to play in future energy systems is gradually becoming evident /1...8, 10, 12, 16...30/. More particular, there is renewed interest in pressurized hot water storage as a means of storing steam and thus the capability of producing electric energy in peaking steam power stations /31...50/, a method that had been used in Europe already in the late Twenties /51...74/.

Today, the wide-spread application of energy storage in electric power generation is hampered by the lack of large storage vessels suitable for high pressures at reasonable cost. Welded steel vessels are too expensive for this purpose. However, the Prestressed Cast Iron Vessel (PCIV) appears to have the potential of complying with these requirements. Its body consists of the inexpensive cast iron, the load due to the internal pressure is taken economically by the very high strength of the steel tendons, it is inherently safe, it is suitable for high and very high pressures and temperatures, it could be available in very large unit sizes, its parts may be handled and shipped in convenient sizes and weights, erection is easy and requires little or no site welding, and dismantling and re-erection is possible.

The PCIV as such was developed as a nuclear reactor pressure vessel by Siempelkamp Giesserei KG of Krefeld, FRG, during the last few years /75...85/. Its application as an energy storage vessel for steam power plants and gas turbine power plants is under development in cooperation with Siempelkamp /11, 16, 37, 38, 39, 42...46, 50/.

The purpose of the Contract (No. EY-76-C-02-2886.⁺000) let by ERDA was to clarify the possible future role of the PCIV technology for energy storage in thermal power stations from the technical, operational and economic point of view. The Contract and this report - the Final Report under the above Contract - include:

- a brief design guide on the PCIV in its role as a thermal energy storage vessel;
- an investigation into the consequences of using a PCIV instead of the well-known welded steel vessel regarding unitsizes, flow schemes for application as peak load or stand-by steam plant, steam conditions, and modes of operation;
- and a parametric study on the use of PCIV for energy storage in connection with nuclear, coal-fired and solar thermal power stations, including specific cost and its dependence on pressure, superheat temperature and unit size.

2. DEVELOPMENT STATUS OF THERMAL ENERGY STORAGE FOR ELECTRIC POWER

Even if the technology of the Prestressed Cast Iron Vessel (PCIV) is new, thermal energy storage (TES) as such is not. A discussion of the PCIV's possible place in TES should, therefore, start with setting out the basic principles and with a synopsis of the previous experience in the field and the state of the art. This appears all the more timely as steam storage is relatively well documented in the European, in particular the German, literature, but not very well in the US literature.

2.1 Place of TES in energy storage

The problem of balancing electric supply and electric demand is as old as electric power generation itself. In thermal power stations using fuel (fossil or nuclear), fuel storage permits a mode of operation in which the above mentioned balancing is possible within the limitation given by the maximum plant output. In solar power plants as well as in wind, tidal and most hydro power plants, the pattern of supply is predetermined by the variable availability of those energy sources. In such cases, load balancing becomes even more difficult unless the supply pattern and the demand pattern should happen to coincide.

There are two basic possibilities of load balancing (Fig. 1): shaping the supply according to the demand, i.e. load peaking, or shaping the demand according to the supply /46/. The latter possibility normally entails customer energy storage (storage "after meter") which is promoted by means of tariff rate incentives. Load peaking requires energy storage facilities somewhere in the electrical grid, either after generator (e.g. by means of batteries or superconducting magnets, flywheels or pumped storage hydro plants) or before generator as thermal storage which in turn may be before or after the steam generator.

Storage before the steam generator, i.e. feed water storage, is an interesting solution for peak load generation in a steam power station. It can provide peak load to an extent limited to the amount of power to be gained by cutting off all the bleeds and furthermore by the overload capacity of the main turbine/generator set.

In solar steam power plants, this method is not applicable, because it requires the steam generator to be on full load which is not always possible in a solar-powered station. Storage after the steam generator is, therefore, required. The same applies to the case of instantaneous reserve or stop-gap reserve in thermal power stations. For thermal storage after the steam generator, there are again two possibilities: high temperature storage from and to the live steam line, or medium temperature storage from the live steam line, from the reheat line or from the turbine to an intermediate pressure and temperature level.

At the intermediate temperature level (up to approximately 350°C) the working fluid (H_2O) may be used as energy storage medium. However, at the high temperature level (above 350°C), the use of H_2O becomes uneconomic and high temperature storage media without phase change (solids, liquid metals, HITEC, or organic liquids) or with phase change (molten salt) are to be used (Table I). Storage vessels in this case are unpressurized, but may require stainless steel or other corrosion-resistant material. Also heat exchangers for charging and

discharging are required, and possibly scraping devices on the heat exchanger tubes in order to improve heat transfer /10, 14, 20, 21/. A combination of the intermediate temperature storage system and the often proposed high temperature storage system may be of particular advantage.

There are several specific tasks to be fulfilled individually or jointly by TES for electric power generation:

- (a) peak load coverage by means of thermal base load plants in conjunction with TES (peak shaving proper, but also load leveling, load following, base smoothing);
- (b) coverage of a demand pattern (peak, medium, base) by a primary energy source with systematic or stochastic variations, e.g. industrial waste heat or the diurnal variation of solar radiation, or its short-term variation due to clouds passing-by;
- (c) instantaneous reserve for power plants;
- (d) quick start-up of turbomachinery.

Typical load durations are connected with the above applications (a) through (d). For peak load or demand pattern coverage (a + b), daily variation is typical, that means load durations of the peaking system of up to several hours but less than 24 hours, usually up to 6 hours; in some cases also weekend duties, with charging of the storage system during Saturday and Sunday, and discharging Monday through Friday during peak hours. For instantaneous reserve and quick start-up (c + d), short term storage is typical, with load durations of up to 2 hours, usually not more than 30 minutes.

The advantages of TES for the above tasks may be summarized as follows:

- use of base load fuel (nuclear, coal) and therefore low fuel cost; no load cycling losses, no thermal fatigue (a),
- utilization of excess energy which otherwise would be wasted (b),
- transformation of cheap off-peak power into valuable peak power (a + b),
- ensuring uninterrupted supply of power or steam (c),
- very quick start-up of thermal peak load plants (e.g. gas turbines) without sacrifice of their life time (d).

2.2 Basic principles of TES for electric power

Fig. 2 shows the two basic principles of TES. In the arrangements as per Figs. 2A and 2B, a storage medium of constant specific energy (2A: solid; 2B: liquid) is taken out of the storage vessel. In Figs. 2C and 2D, the storage medium always remains in the storage vessel but it is heated up during the charging period (specific energy is increased) and cooled down during the discharge period (specific energy is decreased). The storage medium may again be solid (2C) or liquid (2D) or it may cycle between the liquid and vapor state or between the solid and liquid state (latent heat storage). Hot water storage is of particular importance for steam power plants since H₂O-steam may be used directly, without heat exchanger, as working medium. However, storage temperatures above 100°C require the use of pressure vessels.

Basic types of hot water accumulators are depicted in Fig. 3 /23/.

Hot water is taken from the uppermost part of the displacement storage vessel (Fig. 3A) during discharging and supplied to during charging; cold water is supplied to the lowermost part during discharging and taken from during charging. There will be a certain degree of mixing at the interface.

Fig. 3B shows the expansion-type storage system /42/. In the charged condition, the vessel is filled with pressurized hot water. Charging is done by supplying to the PCIV simultaneously and mixing steam of at least the upper temperature and the corresponding saturation pressure and water of any lower temperature. For discharge, hot water is extracted by the expansion of a steam cushion, the steam of which is being generated by the water itself, causing a relatively small temperature and pressure drop. In the discharged condition, the vessel is filled with steam. Specific energy (kJ/kg) is higher in the discharged condition, energy content is lower. The hot water extracted during discharge from storage systems according to Figs. 3A and 3B may be used for generating steam, e.g. by flashing, or it may be used as feed water.

Fig. 3C shows the storage system with internal steam generation (varying pressure accumulator). Steam is generated from the hot water content within the vessel itself during discharge and the bulk temperature is thereby reduced. Charging is done by introducing saturated or superheated steam into the water content. Fig. 3D shows a vessel of this type in a vertical arrangement. If there is a substantial amount of throttling during operation, the temperature of the throttled steam is considerably reduced and superheating in a superheater tube bundle inside the storage vessel is possible. This is shown in Fig. 3D as an alternative by dashed lines.

Storage schemes producing steam during discharge and using steam for charging are traditionally called "steam storage" schemes although the storage contains mainly hot water in the charged condition.

Fig. 4 shows the specific storage capacity of varying pressure steam storage expressed as kg steam produced for a given pressure reduction per m^3 initial water volume and per m^3 initial steam volume /49/.

Fig. 5 shows the specific hot water output of expansion-type storage without recharge, expressed as kg hot water per m^3 storage volume, as a function of upper pressure p_1 . The lower pressure p_2 at discharged condition (full water discharge, no steam discharge) is also given.

2.3 Pressurized air storage

Pressurized air storage may be used for storing mechanical or electrical energy in much the same way for air or gas turbines as TES is used for steam power plants /11, 13, 16/. Pressurized air storage is not based on the storage of internal energy but rather on the storage of pressure energy, similar to the storage of energy in the steam cushion of a varying pressure steam accumulator.

Pressurized air storage by means of the PCIV will be discussed in detail in chapter 7. The basic flow sheets are shown in Fig. 6. Displacement schemes (Fig. 6A) usually employ water as a displacing medium; pressure is thus kept practically

constant. Varying pressure storage may be divided - technologically rather than by principle - into underground storage (salt mine or rock cavern, Fig. 6B) and above ground storage by means of pressure vessels (Fig. 6C).

Fig. 7 shows the storage capacity of a pressurized air storage vessel /11/ as a function of pressure ratio p_2/p_1 and the ratio of thermal capacities of the vessel and of the initial air content W_{Fe}/W_0 , where $W_{Fe}/W_0 = 0$ is the adiabatic case (vessel of zero mass, or vessel with internal insulation, yielding maximum temperature drops), and $W_{Fe}/W_0 = \infty$ is the isothermal case. The dimensionless storage capacity is plotted as ratio of density of air after discharge to air density in the charged condition, ρ_2/ρ_1 . The temperature drop of the pressurized air during discharge is shown as ratio of absolute temperatures, T_2/T_1 .

The use of this diagram is explained by an example. Assuming

$$p_1 = 150 \text{ bar}, p_2 = 9 \text{ bar}, p_2/p_1 = 9/150 = 0.06, \text{ and}$$

$$W_{Fe}/W_0 = 5.0,$$

from the diagram one finds:

$$T_2/T_1 = 0.94;$$

$$\rho_2/\rho_1 = 0.64.$$

Assuming further an initial vessel and air temperature of $T_1 = 20^\circ\text{C} = 293 \text{ K}$,

$$T_2 = (0.94)(293) = 275 \text{ K} = 2^\circ\text{C}$$

is obtained. Initial density at 150 bar, 20°C is 175 kg/m^3 . Therefore,

$$\rho_2 = (0.64)(175) = 11 \text{ kg/m}^3,$$

and the specific storage capacity becomes

$$\Delta\rho = \rho_1 - \rho_2 = 175 - 11 = 164 \text{ kg/m}^3.$$

The varying pressure air storage scheme (Figs. 6B, 6C) entails a temperature rise during charging and a temperature drop during discharging. This has a negative effect on storage capacity as well as storage efficiency. With the large heat capacity and high thermal conductivity of the vessel wall (PCIV !), the negative effect is considerably reduced.

2.4 Comparison of energy storage systems

Apart from TES and pressurized air storage, there are several other methods proposed or already used for storing secondary energy for the purpose of peak load generation /37/.

Hydrogen storage, electro-chemical batteries, fly-wheels and magnetic storage have been proposed lately for the generation of peak power but have not yet been sufficiently developed.

Hydraulic pumped storage plants have been built for many decades as short-, medium- or longterm (weekly or seasonal) storage plants. They do, however, require special geographical conditions, the best sites having already been used.

Underground air storage for gas turbines is a relatively new system. A first plant is being built in the FRG /37/. This method requires special geological conditions (large salt deposits or possibly monolithic rock). Besides, the present air storage gas turbine does require additional gas- or oil-firing. It is, therefore, not a pure storage plant, but rather a symbiosis of a storage plant and a fossil-fired peak load plant.

Several parameters are of importance for a technical and economical comparison of storage systems. Fig. 8 /37, 42/ shows the specific storage capacity - kWh of electric energy per m^3 of storage volume - of several storage systems, namely hydraulic pumped storage (A), air storage with gas turbines (B) and steam storage (C), versus storage pressure (geodetic pressure for hydraulic storage). It turns out that steam storage is highest in specific storage capacity, particularly at low and medium pressure. Whereas hydraulic pumped storage and air storage exhibit low specific storage capacity and do, therefore, require inexpensive storage space, steam (hot water) storage may be economic even with pressure vessels. If hot water is stored in rock chambers as has been proposed /86, 87, 88/, total cost may, possibly be somewhat lower, but that presupposes that peak load as well as base load plant are constructed at a site with suitable geological conditions, i.e. monolithic rock.

Another important figure of merit is storage efficiency (sometimes called: energy storage conversion efficiency /48/). It is defined as the ratio of electric energy generated from the energy storage system during discharge to the electric energy which could have been produced with the charging energy. It is a value which is important for technical as well as economic considerations. In Table II the individual losses are listed that determine storage efficiency. Storage efficiency of steam storage plants will be between 0.5 and 0.95.

For air storage plants with gas turbine, the reciprocal value - the relative energy requirement - rather than storage efficiency should be considered in the first place (it is also listed in Table II). The relative energy requirement in this instance consists of two parts: electric energy for the compressor drive during charging and additional consumption of fossil energy during discharging. Storage efficiencies defined by only one of the two parts of energy consumption are fictitious. If, on the other hand, storage efficiency is related to the sum of both energy requirements, it has to be borne in mind that this is the sum of a thermal energy portion and an electrical energy portion.

Storage efficiency should not be confused with thermal efficiency. Whereas storage efficiency indicates the quality of energy storage, thermal efficiency indicates the quality of the energy conversion process of the main plant. Thermal efficiency defines which part of the energy content of the vessel (with condenser temperature as a basis) may be converted into electric energy; contrary to storage efficiency it relates different forms of energy and therefore does not mean much for judging the quality of a TES installation.

2.5 Lines of past development

The history of TES spans a whole century. Its start may be pinpointed with a US patent for a steam storage system of the varying pressure type granted to

Mac Mahon in 1873 /51/. The development of the present-day storage scheme of the varying pressure type began with the German patent granted to Dr. Ruths in 1913, the specification of which had already indicated its use in power generation. The first installation was built 1920 in Malmö, Sweden /52/, the largest installation 1929 at Berlin-Charlottenburg with 1.4 MPa (210 psi) pressure, 50 MW electric power, 67 MW-hours storage capacity, using a separate peak turbine /61, 63, 64/. This plant is now - after 48 years - still in day and night peaking and stand-by operation for the Berlin island network /74/. Fig. 9 shows the cross-section of one of the 16 riveted pressure vessels of 4.5 m (15 ft.) diameter and 21 m (69 ft.) height, Fig. 10 a picture taken in 1972.

In the Thirties, the varying pressure type steam accumulation was further developed to supply medium-pressure (1.2 to 4 MPa) superheated steam by P.R. Gilli using forged high-pressure (12 MPa) accumulator vessels. This system was based on throttling the working steam and superheating it within the accumulator itself; it was applied to locomotives /89/ and to a stop-gap steam storage plant for the municipal works of Vienna-Simmering /68, 69, 70/.

The first constant-pressure (displacement) storage patent was granted to Druid-Halpin in 1893 /51/. It concerned a storage system for increasing boiler capacity by supplying the boiler with stored saturated feed water instead of cold feed water. This system was further developed by Marguerre in 1924 /60/ by combining it with the regenerative feed water heating systems using a displacement-type storage vessel and a turbine with overload capacity. It was applied for the first time to a large power station in the Mannheim municipal works.

Steam storage installations have also been built for aircraft catapults /90/. Even the use of steam rockets has been proposed and tested /91/. A large number of steam storage installations (after World War II with welded steel vessel), mostly according to the varying pressure principle, have also been built for industrial power plants and for supplying industrial process steam /71, 73/ such as required in the chemical and food industries. One of the major applications is the steel industry. Beginning in the Fifties, Basic Oxygen Furnaces (BOF) were and are being built all over the world, many of them with waste heat boilers whose intermittent steam generation requires the installation of steam accumulators /92, 93/ in order to use the steam in an orderly fashion for power generation or process heat purposes (Fig. 11, /93/). A large number of such plants was built by Wagner-Biro AG of Austria and their licensees, some of them in the US (e.g. at Great Lakes steel works, Fig. 12).

Displacement type vessels have also been built for district heating systems /66, 94, 95/, e.g. in Munich, FRG, and proposed for combined power and sea water desalination plants as hot brine energy storage /18/.

The economic incentive for integrated TES power plants became more pronounced with the advent of the nuclear power plant with low fuel, high capital cost. This was discussed by Marguerre and others in the late Fifties /86, 96, 97, 98/. At that time, however, the nuclear power plants were not yet competitive and they were designed for and operated at base load anyway.

These conditions had changed in 1970 when nuclear power had become competitive and the point of time when part of the nuclear capacity would have to participate in load-following was already approaching in some countries. At that

time, the nuclear peak load plant, preferably with separate peaking turbine of constant inlet pressures, superheat accumulators, multi-pressure condenser /99, 100/ and natural-draft peaking cooling-tower and with minimum interference with the base load plant was developed /31...42/.

Later on, somewhat different systems for conventional power plants with turbine surplus capacity were studied. Application of such a concept to future solar steam power plants has been described in 1976 /46/. At about the same time, an investigation into the possibilities of retrofitted thermal energy storage (RTES) for peaking by means of existing fossil-fired and nuclear power stations was published by Bechtel Corporation /48/. Without reasonably priced high pressure storage vessels being immediately available, their work is based on the use of hot oil TES with charging/discharging heat exchangers. Considering the limited number of plants with generating margin and the large down-time required for the installation of a TES plant of the proposed type, the investigation came to the conclusion that RTES is a concept of limited applicability whereas for grassroot plants the concept of TES as such continues to hold promise.

In 1974, at the Detroit World Energy Conference /37, 38/ and almost simultaneously at the Stuttgart VDI Meeting on the storage of secondary energy /39/, the use of the PCIV for TES was proposed for the first time and a preliminary design presented. The advantages of the PCIV for TES applications (larger unit sizes and lower cost than welded steel, higher optimum pressure, inherent safety, etc.) were pointed out.

However, the flow sheets developed originally for large numbers of medium-pressure welded steel storage vessels had to be revised. Completely new flow sheets suitable for the use of PCIV and mostly based on the use of the new expansion-type storage scheme were, therefore, published in 1975 /42/ for separate peaking turbine sets, and in 1976 for overloadable turbine sets using the feed water storage scheme /44/ or the more recent combined feed water and steam arrangement /46/. The expansion-type feed water and steam storage scheme is presented for the first time in this report.

In 1976, the application of PCIV to very high pressure air storage for the quick-start-up of open cycle gas turbine sets /11/ was proposed and also the use of the PCIV for pressurized high-temperature, sensible heat storage by means of solids (heat-resistant brick checkerwork or cast iron blocks) with a gaseous heat transfer medium (helium in this instance) for closed cycle gas turbines of future Brayton cycle solar power plants /29, 30/.

In this report the use of PCIV for TES is investigated in somewhat more detail. Main emphasis is placed on the steam storage application since this application has the potential of realization in the very near future. A reference design of a large PCIV for TES is developed, stress analyzed, and costed. The flow sheets suitable for a TES peak power plant with PCIV steam and/or feed water storage, and with different base load plants - nuclear, solar, coal-fired - are discussed. In a parametric study, an optimization of the main parameters is carried out, and the capital cost of TES peak load plant and the cost of TES peak power is estimated for a variety of conditions.

3. GENERAL DEVELOPMENT STATUS OF THE PCIV

3.1 Design Philosophy

Until recently pressure vessels for medium and high pressures were made of steel (riveted, since the Thirties welded). For special applications, in particular at high pressures, prestressed multi-layer steel pressure vessels have been and are being used occasionally in the chemical industry. The concept of prestressed steel vessels with prestressing cables instead of sheet metal and with an inner temperature-controlled membrane wall is now being developed as the MWPV (Membrane Wall Pressure Vessel) by the Deutsche Babcock AG of Oberhausen, FRG; whether this concept will be suitable and economical as a thermal energy storage vessel, remains to be seen.

In the late Fifties and the Sixties the need of large pressure vessels for gas-cooled reactor technology had emerged. For this purpose Prestressed Concrete Pressure Vessels (PCPV) were built in France and Great Britain /101/ and later on one plant each in the US and in Germany. The reasons for switching from welded steel to the PCPV were mainly the following:

- safety, i.e. no crack propagation over the entire load carrying cross-section is possible due to the multitude of independent and redundant tendons;
- possibility of site erection;
- shifting the technological limitations toward higher pressures and larger sizes.

The PCPV utilizes three different components for the three basic functions:

- (1) tendons consisting of high tensile strength wires which are prestressed to such a degree that only compressive stress occurs in the vessel;
- (2) a reinforced and cooled concrete structure to carry this compressive stress, to define the shape and to contain all penetrations required.
- (3) a liner as a means to ensure leak tightness of the vessel.

Disadvantages of the PCPV concept are:

- relatively low compressive strength (50 MPa = 7000 psi) and almost zero tensile strength of the concrete;
- no dimensional stability, i.e. shrinkage and creep of the concrete;
- possibility of pressurized crack, i.e. build-up of internal pressure in the concrete wall up to outer diameter in case of liner leaks;
- cooling system required since concrete of high compressive strength has to be kept at temperatures lower than 100°C.

In the late Sixties a new prestressed vessel concept was proposed by the Siempelkamp Giesserei KG, Krefeld, FRG: the Prestressed Cast Iron Pressure Vessel (PCIV) utilizing cast iron blocks of suitable size and weight, machined as necessary, with keys to ensure their relative position. Although 1 m³ cast iron

is more costly than the same amount of concrete, for this new concept a number of important technical and economical advantages are claimed:

- high compressive strength of cast iron (about 800 MPa which is 16 times more than concrete) and relatively high tensile strength (200 MPa) and high bending strength (400 MPa);
- high degree of dimensional stability;
- much lower wall thickness and therefore no catastrophic consequences of a pressurized crack;
- lower weight, lower foundation cost;
- no cooling system required;
- hydraulically prestressing (and restressing, if required) of axial as well as tangential cable bundles instead of winding individual tendons becomes possible by the high compressive strength of cast iron;
- shorter time of erection due to prefabricated cast iron blocks;
- possibility of disassembly and re-erection if required;
- potential of hot going vessels avoiding internal insulation;
- potential of avoiding a separate liner, i.e. leaktight or sealed blocks and sealed interfaces.

3.2 Materials data and stress limits of cast iron

3.2.1 Properties of cast iron

Materials data of cast iron that are relevant to this investigation are presented in Table III. The cast iron considered is of the lamellar type (i.e. grey cast iron). Its trade name is SKM-U 84 of the Siempelkamp Giesserei KG, Krefeld, FRG. It corresponds to specification "GG-20" of the German standard DIN 1691 /102/.

In GG-20, the "20" means the minimum tensile strength (in metric units) of test samples that have been cast separately with a diameter of 30 mm. Such a sample is equivalent to a large casting with 15 mm wall thickness. For larger wall thickness the minimum tensile strength is lower. However, for cast iron according to the Siempelkamp standard SKM-U 84 such a reduction of minimum tensile strength is not specified; the full value of 20 kp/mm² is guaranteed for the whole cross-section up to 300 mm wall thickness that is about the largest wall thickness to be used in a PCIV.

Apart from GG-20, there are other cast iron types specified in DIN 1691, partly with higher tensile strength, for instance GG-40 with twice the value of GG-20. However, it is not the tensile strength, but rather the compressive strength that is of interest for PCIV applications. As is shown in Fig. 13 /78/, compressive strength increases as the square root of the tensile strength. Compressive strength of SKM-U 84 cast iron is about 800 MPa, that is about 4 times the tensile strength of GG-20 which is 20 kp/mm² $\approx$ 200 MPa. This high value of the compressive strength is due to the fact that the notch effect of the graphite lamellas is more pronounced for tensile than for compressive stresses.

Bending strength of cast iron is about twice the tensile strength. This is partly due to the stress-strain relation being strongly non-linear; The Young's modulus of cast iron is stress-dependent, and it is smaller for high tensile stresses where strain is therefore larger. This means that there are less pronounced stress peaks in the outermost region of a cross-section subjected to bending than would be the case with constant Young's modulus. By the same token, neutral axis is not in the center but is moved towards the compressed side. Compressive stress is therefore higher, and tensile stress lower than is assumed in a calculation with constant Young's modulus.

Shear strength of cast iron is much higher than half of the tensile strength as is generally the case for materials with low ductility.

Regarding alternating stresses, cast iron is also different from materials more common in mechanical engineering. This is shown in a Smith plot (Fig. 14, /103/). In this diagram the mean stress (time-wise) is shown on the abscissa, and maximum and minimum values on the ordinate. Permissible deviations from mean stress may be found against a line with the slope 45° . It is evident that large deviations are permissible only in the compressive region. The values of the Smith plot are important for the hot going PCIV which can possibly be subjected to a large number of temperature cycles. This is why Siempelkamp is presently carrying out a test program with the aim to provide more exacting and improved values of alternating stresses for the cast iron type to be used for the PCIV and for the numbers of cycles to be expected in a PCIV. First results of these test are due during 1977; they could not, therefore, already be used for the stress analysis presented in this report.

In part (e) of Table III, various other properties of cast iron are listed. As mentioned, Young's modulus is stress dependent in the tensile region.

3.2.2 Stress limits according to ASME III and VIII/2

For ferritic carbon and low alloy steel, the "design stress intensity" S_m is used as a limit for primary stresses in the ASME Code, Section III /104/ and Section VIII, Div. 2 /105/. It is defined as:

2/3 of yield strength, or

1/3 of ultimate tensile strength

whichever is smaller. This procedure applies to primary tensile as well as to compressive stresses. In case of combined stresses, one has to take the "stress intensity" S (according to the maximum shear stress hypothesis) which is equal to the largest of the three values

$$S_{12} = (\sigma_1 - \sigma_2)/\sqrt{2}$$

$$S_{23} = (\sigma_2 - \sigma_3)/\sqrt{2}$$

$$S_{31} = (\sigma_3 - \sigma_1)/\sqrt{2}$$

σ_1 , σ_2 and σ_3 are the principal stresses. In case of three-dimensional tensile stress, the algebraic sum of the three principal stresses ($\sigma_1 + \sigma_2 + \sigma_3$) shall not exceed four times the tabulated value of S_m .

This method, obviously, cannot be applied to cast iron, for the following reasons:

- Steel obeys Hook's law up to yield strength, cast iron not at all. There is no defined yield strength and Young's modulus is stress-dependent.
- As already mentioned, compressive strength is higher (about 4 times) than tensile strength. The hypothesis of maximum shear stress is therefore not applicable. The ASME Code III /104/ and VIII, Div. 2 /105/ in its present state should not be applied to highly stressed cast iron structures.

3.2.3 Stress limits according to ASME III/2

Apart from ASME Codes III /104/ and VIII, Div. 2 /105/, there is the new ASME Code, Section III, Div. 2: Code for Concrete Reactor Vessels and Containers /106/ which applies to the PCPV and might therefore be applied in its essence to the PCIV. It defines design stresses for concrete as follows (Table CB-3412-1):

- tensile stress: $f_{ct} = 0$, i.e. any primary stress must not be in the tensile range;
- compressive stress: $f_{cc} = 0,4 \sigma_{cuc}$, i.e. primary compression stress must not be higher than 40 % of ultimate compressive strength (except for triaxial compressive stress where higher values are permissible).

3.2.4 Stress limit philosophy for cast iron

For multi-axial stresses, there is a graphical method for defining stress limits, the Mohr plot. Direct stress is plotted on the abscissa, the corresponding shear stress on the ordinate. The combinations of direct and shear stress are given by circles whose centers are on the abscissa and whose intersections with the abscissa define the principal stresses. The largest radius of the circles corresponds to the maximum shear stress. According to the more general Leon hypothesis the multitude of all that circles which describe a permissible combined stress condition is embraced by a parabola whose axis coincides with the abscissa.

Fig. 15 shows the fundamental differences between steel and cast iron in the Mohr/Leon plot. For steel, the head of the parabola (triaxial tension) is far in the tensile region. Compressive strength is only marginally higher than tensile strength; shear stress is relatively low; the maximum shear stress hypothesis is applicable. For cast iron, the head of the embracing parabola is near the center; shear stress is high, compressive strength is high; the parabola has a large angle of aperture; the maximum shear stress hypothesis is not applicable. For cast iron, the hypothesis of maximum principal stresses - which is based independently on maximum tensile strength and maximum compressive strength - should be used rather than the maximum shear stress hypothesis.

3.2.5 Stress limits used

The following stress limits are used by Siempelkamp and are also being used in this investigation:

- tensile strength: $f_{ct} = 0$ for primary tensile stresses;
- compressive strength: $f_{cc} = \sigma_{cuc}/4$ for primary stresses;
 $f_{cc} = \sigma_{cuc}/3$ for secondary stresses.

The above stress limits are more conservative (0.25 against 0.4 σ_{cuc}) than the ones of ASME Code for PCPV /106/ although scatter of experimental values is much less for cast iron than for concrete. No higher stress limits are used for three-dimensional compression. This is partly due to the fact that the (compared to the PCPV) lower wall thickness of the PCIV will not lead to truly three-dimensional stresses to the same extent as the very large wall thickness of a PCPV.

3.3 Materials data and stress limits of cables

Material data for the cables are presented in Table IV. A high-tensile strength material is used, with very high yield strength due to a considerable degree of cold working during production. The tendons are subjected to one-dimensional tension.

Design stresses are defined according to ASME III/2 /106/ for design conditions as being:

- 80 % of σ_{sut} , or
- 90 % of yield strength σ_{sy} or $\sigma_{s0.01}$

whichever is smaller. For the failure condition, the full value of σ_{sut} may be used.

Relaxation is an important consideration. In part (e) of Table IV strains are specified that are experienced by tendons after a specified time under operational prestressing to 0.5 to 0.85 of σ_{sut} , which is in the range of prestressing.

3.4 Basic stress analysis

3.4.1 General

According to the ASME Code /104, 105, 106/, stresses may be either primary, or bending, or secondary, or local peak stresses. For the PCIV, stresses may be classified into stress categories as follows.

(a) Cast iron

Primary stresses:

- pressure-induced membrane stresses;
- stresses due to prestressing.

Primary bending stresses:

- pressure-induced bending stresses in flat plate;

Secondary stresses:

- bending stresses between cylindrical part and transition part of flat plate;
- thermal discontinuities due to an axial temperature gradient in the cylindrical wall;
- thermal discontinuities due to different rates of temperature change in cylindrical part and flat plate;
- thermal discontinuities due to a linear temperature gradient in the flat plate (axial direction).

Local peak stresses:

- stress concentrations at notches, holes, transitions, etc.;
- local radial loads (shoes);
- thermal peak stresses due to a radial temperature gradient;
- thermal peak stresses due to a non-linear temperature distribution in the flat plate;
- thermal peak stresses due to local temperature peaks (e.g. hot charge line).

(b) Tendons

Primary stresses:

- due to prestressing;
- due to internal pressure;
- due to temperature difference to cast iron structure (e.g. hot going vessel, cold going cables).

Secondary and local peak stresses:

- none.

3.4.2 Design stresses

According to ASME III /104/ and VIII/2 /105/, the following limits apply to design stress intensities at operating conditions:

- Primary stress intensity: $1.0 S_m$
- Bending stress intensity: $1.5 S_m$
- Secondary stress intensity: $2.0 S_m$
- Local peak stress intensity: $3.0 S_m$

All stress categories - including local peak stresses - are to be considered for fatigue analysis.

3.4.3 Design base

There are basic differences in stress distribution and in behavior during failure conditions of conventional pressure vessels made up of homogeneous material and prestressed pressure vessels with a prestressing system that takes up internal pressure, stresses the main body, and reacts with it. Fig. 16 shows the reactions for the cylindrical part.

During prestressing the prestressing system is strained and the main body compressed until in the prestressed condition forces are in equilibrium. During the condition of internal pressure, the main body remains in the compressed condition, and tensioning of the prestressing system is slightly increased. There should be a safety factor F_g against the main body losing its compressive state (i.e. tension in the main body if it consists of rings or gapping of the body if it consists of sectors which is usually the case for larger PCIV). Gapping of the main body leads in the case of a PCIV without liner to leaking and subsequently to a loss of internal pressure; in case of a PCIV with liner the liner will lose its integrity as soon as the tendons begin to yield and leaking will also occur; a further gapping (dotted lines in Fig. 16) is not possible (i.e. the vessel is inherently safe). Fracture of the tendons should be sufficiently far from gapping (safety factor F_i).

In short, there are the following basic safety factors:

F_g	against gapping, i.e. ratio of gapping pressure to design pressure;
F_i	factor of inherent safety, i.e. ratio of pressure necessary for tendons failure to gapping pressure;
$F_f = F_g F_i$	against tendons failure, i.e. ratio of pressure necessary for tendons failure to design pressure.

According to the practice at Siempelkamp, the following values of the safety factors are chosen for the investigations:

$$\begin{aligned}F_g &= 2.0 \\F_i &= 1.5 \\F_f &= (2.0)(1.5) = 3.0\end{aligned}$$

Recently, a somewhat more advanced design base is used by Siempelkamp. It consists of:

- $F_g = 1.5$
- tendon stress at design pressure limited to $\sigma_{s0.2}/1.5$.

Depending on relative elasticity (ratio of cross-sections and of Young's moduli of tendons and cast iron main body) as well as on ratio of yield strength to ultimate tensile strength of wires, with the above more advanced assumptions safety against tendons failure becomes approximately $F_f = (1.5)(1.5) = 2.25$.

It should be noted that gapping safety F_g need not necessarily be exactly the same for axial and tangential cables. For instance, F_g may be somewhat lower for the axial direction. In this way, the mode of gapping may be defined.

3.4.4 Cylindrical wall

The interaction between tension of the prestressing system, compression of the main body, and internal pressure has been described in the previous paragraphs for a radially and tangentially stressed ring element using the scheme of Fig. 16. In the undisturbed cylinder, there is one additional stress: axial prestressing. If the utilization of the axial tendons is taken the same as for the tangential ones, it follows that cable density in the axial direction is only half the density in the tangential direction.

Stress in the cylinder is also only one half in the axial direction of what it is in the tangential direction. This is identical to a non-prestressed cylinder. Therefore, the ratio of axial and radial stress is also identical to the ratio in a non-prestressed cylindrical pressure vessels.

The reactions of axial forces on radial strain due to Poisson's ration will be dealt with later (para. 4.2), also the effects of discontinuities (para. 3.4.6).

3.4.5 Flat plate

Similar to the stress-strain diagram (Fig. 16) for the radial strain of the ring element, a simplified stress-strain diagram may be drawn for the flat plate (Fig. 17). The following steps are considered:

- flat plate with prestressing;
- prestressing;
- prestressing and internal pressure.

Stresses and strains are different in the outer and inner flange of the flat plate. The inner flange is under compression which is increased due to internal pressure, whereas compression in the outer flange is reduced by internal pressure. Gapping occurs at the outer flange; at the same time the inner flange experiences maximum compression. This is fundamentally different to the conditions in the cylindrical wall where maximum compression occurs in the depressurized condition.

This has an effect on the safety philosophy. For the cylindrical part, interactions between stress and strain are more or less equal in the whole cross-section of the wall, and a gap will develop over the whole wall thickness. Internal pressure will be reduced by this gap acting as a safety valve if it is assumed that the liner breaks at or near the gap. The system is inherently safe. At the flat plate, on the other hand, gapping occurs primarily at the outer flange; the inner flange will be compressed even more.

A further difference between cylindrical part and flat plate concerns the shear forces at the block boundaries: In the cylindrical part there is no significant shear stress at the block boundaries, at least in the undisturbed part.

In the flat plate, there will be shear forces; they are taken up by friction (by force) on the one hand and by the keys (by shape) on the other.

Regarding tangential cables at the flat plate, the optimum arrangement consists not in an equal thickness of the cable bundle because this leads to an unnecessary high prestressing due to the reactions to internal pressures. It is better to shape the cable bundle in such a way that the outer portion gets the larger thickness. For such a wedge-type cable bundle, there is less compression stress during operation at the inside and at the same time gapping is shifted to higher internal pressures.

Further savings of cast iron and cables may accrue for a hollow plate girder type design if only the outer and the inner flange are designed with large wall thickness and the intermediate horizontal and vertical walls are designed only with minimum wall thickness as required by the casting procedure. Such a hollow design will require much less cast iron weight, compared to a solid plate of the same stiffness. Tendons will in this case be arranged mainly at or near the outer and inner flange.

A third improvement is the "step design" of the flat plate. Height of the plate is largest in and near the axis of the vessel, it is reduced stepwise with increasing radius. There is a cable bundle at each step which supplies the necessary compression locally and directly without carrying it through the whole structure. There will be a considerable saving of cast iron as well as of cables (due to the smaller radius).

It should be borne in mind that the possibilities of saving weight and cost of the flat plate (wedge-type cable bundle, flange stressing, and step design) do not exclude each other. They may be realized simultaneously which represents the most economic design of the flat plate.

3.4.6 Discontinuities

Apart from internal pressure and prestressing there are the well-known discontinuity stresses at the boundary between flat plate and cylindrical part, caused by the different stiffness of the two system. Such stresses are secondary stresses according to ASME categories.

With increasing internal pressure the radius of the cylindrical part will markedly increase whereas the radius of the flat plate will practically not. It will, on the other hand, bend towards the outside. There will thus be additional stresses and bending moments created in the transition zone. Stresses due to the bending moment will not be very large in the flat plate due to its stiffness; it is the less stiff cylindrical part that will have to compensate for the angular movement of the plate boundary. The rim of the flat plate may therefore be considered to be somewhere in between the freely supported and the fixed condition.

It is difficult to find the optimum shape of the cylinder/plate transition zone. In any case, the corner part should not be sharp but rather rounded and it should be part of the plate rather than the cylinder. The boundary between the plate casting and the neighboring cylindrical casting should be somewhere (a few wall thickness) in the cylindrical part. For the design and assessment the work of Stockmann and Decock /107/ should be considered.

3.5 The PCIV as nuclear reactor vessel

The PCIV concept was first applied to the nuclear field. In fact, the concept of the PCIV was developed as an alternative to the Prestressed Concrete Pressure Vessel (PCPV) of the German High Temperature Reactor THTR of 300 MW electric power now being built at Schmehausen, FRG (Fig. 18, /84/).

This is not incidental. Nuclear reactor pressure vessels belong to the largest high-pressure vessel in terms of pressure times volume. The THTR reactor vessel has a working pressure of 40 bar and has an internal volume of 3040 m^3 and thus a pressure-volume of $122\,000 \text{ bar m}^3$. The largest reactor vessel of the British Magnox type, Wylfa, had 27 bar and $13\,100 \text{ m}^3$ and thus $354\,000 \text{ bar m}^3$. the largest French gas-cooled graphite-moderated reactor, Bugey, had 45 bar, 8780 m^3 and $377\,000 \text{ bar m}^3$. The reference TES vessel (para. 4.3) has $p_0 = 60 \text{ bar}$ operating pressure, an internal volume $V = 8000 \text{ m}^3$, and therefore $p_0 V = 480\,000 \text{ bar m}^3$. From what has been said before it is obvious that the economic advantages of the PCIV are most pronounced if pressure-volume is high.

Besides, it turned out that the biological shielding requirements could be met by the cast iron alone /78, 82/. So, there was no need for a separate biological shield as would have been in the case of a welded steel vessel (of lower wall thickness). For very stringent requirements, some of the caves of the hollow cast-iron blocks may be filled with a suitable shielding material. This was investigated in detail by Hochttemperatur-Reaktorbau GmbH (HRB) /82/.

A rubber-lined test vessel (Fig. 19) was built by Siempelkamp Giesserei KG at their Krefeld, FRG, works during 1972. The cylindrical part consisted of 3 rings of 12 blocks each. This test vessel was a 1 : 7.5 scale model corresponding essentially to the design for the THTR pressure vessel; it was cold tested at 9 MPa and behaved very well. The first publications on the PCIV concept in the open literature appeared from 1973 /75...85/, amongst them an evaluation by the German Battelle Institute under contract with the government of Nordrhein-Westfalen /78/.

The PCIV concept was furthermore considered by Siempelkamp /84/ and by Westinghouse /108, 109/ for the proposed large high-temperature process heat reactors of 3000 MW thermal power.

The excellent safety prospects of the PCIV have also lead to a PCIV design for light water reactors. Fig. 20 shows the general arrangement of a Boiling Water Reactor PCIV design for a future plant of 1450 MW electric power per unit. The upper plate consists of an inverted vault structure; removable struts transmit the compression forces to the cylindrical wall. Tangential cables are wound around the cast iron blocks; axial cables are situated within the cast iron blocks (Fig. 21). The steam pipe penetration consists of a sleeve socket fixed to the cast iron wall by studs, the cold water inlet pipe which is welded to the sleeve, the inner hot pipe, a flow limitation device, and the insulating plus cooling elements /84/.

The design is conservative in the sense that the PCIV walls are kept at a temperature of 650°C by means of a pressurized wire-mesh thermal insulation and a cooling system in a pressurized water gap between the liner on the cast-iron blocks and the thermal insulation (Fig. 21).

The PCIVs described so far - except the test vessel - are designs but have not yet been built. The PCIV shown in Fig. 22, however, is a PCIV that has been ordered from Siempelkamp for the THTR Schmehausen where it is to act as a control rod drive pressure vessel [11, 43, 85]. After being tested at the Krefeld facilities of Siempelkamp it will be shipped to site during autumn 1977. Its internal diameter will now be 2.0 m, its internal height 5.6 m, the internal volume 17.5 m³, its design pressure 23 MPa (230 bar, 3300 psi), the cylindrical wall thickness 240 mm, the total weight 150 tonnes. This is a nuclear vessel and it is being approved by the German authorities (TUV).

3.6 Code situation

For the large-scale introduction of PCIV for TES, the question of Code compatibility will be of importance. PCIV is new technology and it is therefore not surprising that it has not as yet received proper attention by the ASME Code. There is presently no PCIV ASME Code. Whether and to what extent existing ASME Codes may be applied, has already been discussed in previous paragraphs (3.2.2 and 3.2.3).

In conclusion the following may be said:

- ASME Codes for steel pressure vessels - i.e. ASME III /104/ and ASME VIII/2 /105/ - are not applicable to prestressed systems on the one hand and they are not applicable to cast iron on the other. In several codes of other countries, the use of cast iron for pressure-bearing parts is mentioned and the use of cast iron is restricted to low pressures and small volumes. Such restrictions, of course, do not apply to the PCIV which is a prestressed system and, as such, is based on the high compressive strength of the cast iron rather than the tensile strength.
- Parts of the ASME Code III/2 for concrete reactor vessels /106/ may be applied to the PCIV. This concerns the basic philosophy on the values of permissible design stresses for cast iron in analogy to concrete (although at present a more conservative safety factor of 4.0 is applied to cast iron against 2.5 according to Table CB-3421-1 of ASME III/2 for concrete). It also concerns the tendons where Table CB-3421-2 of ASME III/2 is fully applicable to the PCIV. Also for the liner, rules for permissible stresses and strains (Table CB-3700-1) may be applied to the PCIV.

4. BRIEF DESIGN GUIDE OF PCIV FOR PRESSURIZED HOT WATER

4.1 General design features

Using the PCIV as an energy storage vessel permits new and interesting features as compared to the PCIV as a nuclear reactor vessel and at the same time has several important advantages against the use of the welded steel vessel as energy storage vessel, in particular pressurized hot water vessel:

- The PCIV as a pressure vessel for gas-cooled nuclear reactors has to cope with height (length) to diameter ratios L/D around 1.0 which are very far from the optimum. In energy storage applications, the optimum L/D ratio may be chosen; it will be above $L/D = 4.0$. As is discussed later (para. 6.2.2) in detail, the optimum L/D is determined mainly by the decreasing cost of flat plates and the increasing cost of thermal insulation with increasing L/D .
- For TES applications, very large unit size are required; these are most suitable for the use of PCIV.
- For TES applications in steam power plants, operational pressures which are of interest are in excess of those utilized for gas-cooled reactors vessels and far in excess of those economically possible with welded steel vessels.
- Penetrations of large pressure vessels in general and prestressed vessels in particular are complicated and expensive. Contrary to the PCIV for nuclear reactor applications, the PCIV for TES requires essentially only a charge and discharge pipe penetration of moderate size that may be conveniently located at the center of the flat plates. In this way, there are only few fixed points for the liner which may thus be of the floating design.
- The PCIV is made up of mass-produced units of sizes, shapes and weights that are conveniently transported and may be connected on site essentially without site welding and stress relieving operations.
- Contrary to concrete, cast iron may be used at high temperature. The PCIV for TES may therefore be designed as a hot going vessel by locating thermal insulation outside the main body of the vessel thus avoiding the difficulties of thermal insulation within the vessel. Thermal insulation of a hot going PCIV may be outside the cables (which are in this case hot going, too) or it may be between the main body and the (cold going) cables or shoes as a pressurized thermal insulation.

4.2 Specific stress analysis, steady-state

A specific stress analysis for steady-state conditions will now be performed for the three basic types of the PCIV: cold going, hot going with hot going cables, and hot going with cold going cables. Based on the results of the stress analysis, the reference design will be decided upon and subsequently stress-analysed for transient conditions, in particular regarding transient thermal stresses.

4.2.1 Cold going vessel

The cold going PCIV is similar to the PCPV in that pressurized thermal insulation is provided inside the main body and usually inside the liner, i.e. within the pressurized hot water. Several design possibilities of the thermal insulation exist some of which have been tested for PCPV applications but have not yet been built in an actual plant:

- one or several interspaces filled with pressurized gas whose pressure is accurately controlled so as to be in equilibrium with internal water pressure /110/;
- several interspaces filled with stagnant water /84/;
- porous concrete of low thermal conductivity /111, 112/ or Microtherm in stainless steel cladding either inside the liner or between liner and main body;
- "Isographite" or "Klingerit 1000" with or without cladding (see para. 4.4.3).

The following basic relations apply to stresses σ and strains ϵ :

(a) General

Hook's law with Young's modulus E and Poisson's ratio ν reads for one-dimensional stresses ($\sigma_1 = \sigma$, $\sigma_2 = \sigma_3 = 0$, e.g. tendons):

$$\epsilon_1 = \sigma_1/E \quad (4.1a)$$

$$\epsilon_2 = \epsilon_3 = -\nu \sigma_1/E \quad (4.1b)$$

and for three-dimensional stresses:

$$\epsilon_1 = [\sigma_1 - \nu(\sigma_2 + \sigma_3)]/E \quad (4.1c)$$

$$\epsilon_2 = [\sigma_2 - \nu(\sigma_3 + \sigma_1)]/E \quad (4.1d)$$

$$\epsilon_3 = [\sigma_3 - \nu(\sigma_1 + \sigma_2)]/E \quad (4.1e)$$

For a cylindrical part under internal pressure ($\sigma_1 = \sigma_t$; $\sigma_2 = \sigma_x = \sigma_t/2$; $\sigma_3 = \sigma_r = 0$), tangential strain becomes

$$\epsilon_t = (\sigma_t - \nu \sigma_x)/E = (1 - \nu/2)\sigma_t/E \quad (4.1f)$$

and axial strain:

$$\epsilon_x = (\sigma_x - \nu \sigma_t)/E = (1 - 2\nu)\sigma_x/E \quad (4.1g)$$

(b) Cylindrical part

From equation (4.1a) applied to ultimate tensile strength σ_{SUT} of the cables (which is identical to the design stress at failure conditions, f_{sf}) the maximum elastic strain (fictitious value) of the cables is obtained (Fig. 23):

$$\epsilon_{\max} = f_{sf}/E_s \quad (4.2)$$

Strain of tendons at the gapping condition becomes:

$$\epsilon_g = \epsilon_{\max}/F_i \quad (4.3)$$

Strain at prestressing of the tangential cables:

$$\epsilon_p = \epsilon_g - f_{cc}(1 - \nu/2)/E_c \quad (4.4)$$

From a balance of forces the thickness of the tangential cable system (considered as a solid ring) is obtained:

$$s_s = F_f R p_i / f_{sf} \quad (4.5)$$

where p_i is the internal design pressure that is the sum of the operating pressure p_o , the gravity head as applicable and a safety margin. In a similar manner the thickness of the main body under compression due to prestressing:

$$s_c = \epsilon_p s_s E_s / f_{cc} \quad (4.6)$$

and the cross-section A of the axial tendons with vessel radius R and internal design pressure p_i :

$$A = F_f R^2 \pi p_i / f_{sf} \quad (4.7)$$

(c) Circular flat plate

The circular flat plate will be designed with a safety factor against gapping (Fig. 17):

$$p_g = F_g p_i \quad (4.8)$$

Permissible bending stresses in the flat plate (Fig. 17) are:

$$\sigma_b = f_{cc}/2 \quad (4.9)$$

If the plate is considered to be freely supported, radial bending becomes according to ASME III /104/ with plate thickness s_{cp} :

$$\sigma_r(r, x) = p_i \frac{3x}{4s_{cp}} [(3 + \nu)(R^2 - r^2)] \quad (4.10)$$

and tangential bending:

$$\sigma_t(r, x) = p_i \frac{3x}{4s_{cp}} [(3 + \nu)R^2 - (1 + 3\nu)r^2] \quad (4.11)$$

where x is the longitudinal distance from the midplane and r the radial distance from the center of the plate.

Maximum bending stress occurs at the center of the plate ($r = 0$) at its surface ($x = s_{cp}/2$) and becomes with $\nu = 0.28$:

$$\begin{aligned} \sigma_{bmax} &= \sigma_r(0, s_{cp}/2) = \sigma_t(0, s_{cp}/2) = p_g(R^2/s_{cp}^2)(3.28)(3/8) = \\ &= 1.23 p_g R^2/s_{cp}^2 \end{aligned} \quad (4.12)$$

The required plate thickness therefore is given by

$$s_{cp} = \sqrt{1.23 p_g R^2/\sigma_b} \quad (4.13a)$$

The solid plate may be replaced by a girder type plate with an equivalent resistance to bending. If h_{po} stands for the outer and h_{pi} for the inner height of the girder type plate, the equivalent solid plate thickness is

$$s_{cp \text{ eq}} = \sqrt{(h_{po}^3 - h_{pi}^3)/h_{po}} \quad (4.13b)$$

The cables of the flat plate have to produce a tangential stress as high as σ_b at gapping conditions, therefore the thickness of the cable bundle, considered as a solid ring, has to be

$$s_{sp} = F_i (R + s_c) \sigma_b / f_{sf} \quad (4.14)$$

An example of a cold going PCIV is worked out in Table V.

4.2.2 Hot going vessel, cold going cables

The concept of hot going vessel with cold going cables requires pressurized thermal insulation between cables (shoes) and the main body. If the main body is heated up (Fig. 24), prestressing is increased (self-stressing).

The following equations pertain:

- ϵ_{max} as given by equation (4.2)
- Gapping of the hot body:

$$\epsilon_{gh} = \epsilon_{max} / F_i \quad (4.15)$$

- Gapping of the cold body (colder by ΔT), with coefficient of thermal expansion β :

$$\epsilon_{gc} = \epsilon_{gh} - \beta \Delta T \quad (4.16)$$

- Strain due to prestressing, hot condition:

$$\epsilon_{ph} = \epsilon_{gh} - f_{cc}(1 - \nu/2)/E_c \quad (4.17)$$

- During operation with design pressure, the strain of the axial tendons becomes:

$$\epsilon_i = \epsilon_{ph} - (\epsilon_{gh} - \epsilon_{ph})/F_{gh} \quad (4.18)$$

and the corresponding stress in cables is

$$\sigma(\epsilon_i) = E_s \epsilon_i \quad (4.19)$$

- Thickness of the tangential prestressing system, considered as solid ring, i.e. tendons cross-section per m of height:

$$s_s = \frac{F_{gc} R p_i}{f_{sf} \epsilon_{gc} / \epsilon_{max}} \quad (4.20)$$

Cross-check:

$$F_f = \frac{s_s f_{sf}}{R p_i} \geq F_{f \text{ req}} \quad (4.21)$$

$$F_{gh} = F_f / F_i \geq F_{gh \text{ req}} \quad (4.22)$$

The required thickness of the main body is obtained from the equilibrium condition at prestressing:

$$s_c = \epsilon_{ph} s_s E_s / f_{cc} \quad (4.23)$$

According to Siempelkamp practice f_{cc} is taken at operating temperature, which is a little bit conservative as these conditions occur only in a PCIV which has no internal pressure but has still operation temperature. This could be considered as a failure condition.

Safety factors are connected by the following relation, obtained from equations (4.15), (4.16) and (4.18):

$$F_{gc}/F_{gh} = 1 - \beta \Delta T F_i / \epsilon_{max} \quad (4.24)$$

The flat plate will be designed with the same safety factor against hot gapping as the cylindrical part. Therefore

$$p_g = F_{gh} p_i \quad (4.25)$$

Equations (4.9) to (4.14) are also valid for hot going vessels. Example calculations for a hot going PCIV are given in Table VI according to the conservative design base, and in Table VII according to the advanced design base.

4.2.3 Hot going vessel, hot going cables

The concept of the hot going vessel requires main body as well as prestressing system to be located inside the thermal insulation. The following differences between this concept and the cold going PCIV prevail:

- Strength and Young's modulus of cast iron as well as of cables are to be taken at operating temperature.
- Depending on whether the thermal expansion coefficient β of cables is higher or lower than the one of cast iron, there will be a decrease or an increase of the prestressing as compared to the cold condition. Calculation of these effects is analogous to the concept of the hot going vessel with cold going cables; however, the effects will be smaller by about one order of magnitude.
- Transient thermal stresses, too, will be similar to the ones of the concept with hot going vessel, cold going cables (para. 4.5). The problem of transient thermal stresses may, however, be even more difficult.

It should be noted from Table IV that ultimate tensile strength σ_{ut} and yield strength $\sigma_{0.02}$ of the cables at 300°C are considerably less than the values at ambient temperature. This means that already for this reason the concept requires twice the amount of cable weight. A further basic problem of this concept is cable relaxation. There is, at present, only insufficient data available /113...116/ and there seems to be scope for further investigation and development.

Creep limit is usually defined as the value of tensile stress that creates a certain strain after a specified period of time. For pressure-bearing parts at high temperature, usually 1 % strain is permitted after 100 000 or 200 000 hours of operation. Such strains are certainly not permissible for prestressing systems; permissible strain will rather be of the order of 0.1 % or less since the whole prestressing is achieved by about 0.4 % strain (Fig. 22):

$$\epsilon = \sigma/E = 772/210\,000 = 0.00368 = 0.368 \%$$

The steel St 160...180 specified in Table IV for the tendons conforms to the condition of 0.1 % strain only at temperatures of about 100°C. Low carbon alloy steels, such as low Cr or Cr-Mo-alloy steels (0.5 to 2.5 % Cr) or high Cr-alloy ferritic steels (8 to 12 % Cr) may be better in this respect. The same may apply to stabilized austenitic steels /114/. However, there may be the stress corrosion problem.

At 300°C, relaxation will be of such a magnitude that restressing would be necessary after a few years of operation. If hydraulic restressing is considered, the minimum operating time until relaxation reaches the limiting value may be reduced from, say, 200 000 hours $\approx$ 23 years to, say, 50 000 hours $\approx$ 6 years.

4.2.4 Discussion

The results of the specific stress analysis for the three cases considered will now be discussed and the safety factors at various operational conditions analyzed with the aim of arriving at a decision as to which of the three should be chosen as reference design.

The behavior of a cold going PCIV (with $F_f = 2.25$, $F_t = 1.5$, $F_c = 1.5$ according to para. 4.2.1) is shown in Fig. 26. If internal⁹ thermal insulation would be removed, the PCIV would become a hot going vessel and its safety limits would change as follows: With increasing temperature of the main body and $p_0 = 0$, prestressing of the main body would be increased and, finally, the design compressive stress would be exceeded. A simultaneous increase of internal pressure p_0 would, however, reduce the compressive stress. With higher prestressing at higher temperature, gapping limit increases. Ultimate tensile stress of the tendons remains initially constant.

At about 225°C gapping and tendons fracture would occur simultaneously, i.e. the advantage of the prestressed vessel regarding safety - the inherent safety - has been lost at that temperature or at temperatures above.

Fig. 27 shows the pressure-temperature-diagram for the hot going PCIV (with cold going cables), conservatively designed. Safety limits are now extended if compared to Fig. 26. The higher compressive limit of the main body required for the hot depressurized condition has only been possible by a considerably higher cast iron wall thickness. The requirements of inherent safety ($F_t = 1.5$) and safety against cold gapping ($F_{gc} = 1.5$) lead to a large safety factor of the cables ($F_f = 5.79$ against $F_f = 3.0$ for the cold going vessel) and therefore a large cable weight. The saturation curve of H_2O has been drawn into Fig. 27. It is applicable for accumulators with internal steam generation and for expansion-type storage vessels and shows that there is sufficient safety in all operational conditions. Repeated depressurization of the hot vessel may, however, be somewhat dangerous (by exceeding the design compressive stress of the main body).

The combination of high internal pressure and low temperature may, however, occur during normal operation for displacement-type storage vessels with large temperature swings, i.e. for feed water storage. A lower temperature level of 105°C has been chosen in Fig. 27; this temperature is convenient for thermal degassing of the water.

Fig. 28 shows the design limit of a hot going PCIV according to an advanced design base. Fig. 28 is very similar to Fig. 27 but the pressure scale (abscissa!) has been reduced by applying a factor of 0.39. The characteristic values of cables and main body have been reduced accordingly. Gapping safety in the cold condition is now very low ($F_{gc} = 0.58$), that means a pressure test in the cold condition is only possible up to 58 % of design pressure. Only above 125°C may full internal pressure be applied. Similar to the conservative design base, the design is suitable for accumulators with internal steam generation and expansion-type storage vessels, but not necessarily for displacement-type vessels.

The hot going PCIV with hot going cables (all pressure bearing parts hot going as for the welded steel vessel) has already been discussed in para. 4.2.3.

Due to the serious decrease of α_{sut} at operation temperature, the amount of cables is much larger than for cold going cables. Besides, relaxation has to be considered.

Nominal weights of cast iron and of cables (not including margins for non-uniform load distribution of the tangential cables as will be discussed later), according to the simplified considerations of the previous paragraphs, for the three alternatives under consideration for the reference design (8000 m³ net volume, 60 bar = 6 MPa operating pressure) are:

	Casting (tonnes)	Cables (tonnes)
Cold going PCIV	5 600	2 300
Hot going PCIV, cold going cables :		
conservative design base	16 600	5 400
advanced design base	6 400	2 000
Hot going PCIV, hot going cables :	6 400	not available

It turns out that the conservative design of the hot going PCIV with cold going cables according to the conservative design base is very heavy and expensive; however, with advanced design base it requires, due to the self-stressing effect, about the same weight as the cold going vessel if net volumes are compared, i.e. if the increase of the vessel due to internal thermal insulation of the cold going vessel is accounted for.

4.3 Reference design

The hot going PCIV with cold cables according to the advanced design base not only requires the least amount of cable weight of all alternatives considered but also avoids the problems of thermal insulation within the vessel. It is, therefore, chosen as the reference design.

Operational data of the reference design PCIV are given in Table VIII. Nominal values of the capacity of the PCIV used as a steam storage vessel are given in part (c) of Table VIII for both the varying pressure accumulator with internal steam generation and the expansion-type vessel. Nominal values of storage efficiency are presented in part (d) of Table VIII for the two storage types. Although thermal efficiency of the steam or feed water storage system is relatively low, storage efficiency is high even if thermal insulation losses occurring during one or more days of stand-by duty are included.

The reference design PCIV in a vertical arrangement is shown as Fig. 29. The more important design data of the reference design are given in Table IX, a weight estimate in Table X.

An alternative reference design is shown in Figs. 30/31. This is a PCIV arranged horizontally with several supports along the axis. The support in the middle will be a fixed one. This horizontal design is suitable for expansion-type storage, possibly also for internal steam generation, but not for displace-

ment type storage vessels since there would be too much mixing of hot and cold fluid. The design has several advantages:

- Axial tendons may be located inside the tangential ones as is shown in Fig. 30. The flat plates therefore will be less expensive. Such an arrangement is not feasible for a large vertical vessel because installation of the tendons would require the use of a helicopter.
- Static pressure difference due to the height of water is much lower than in a vertical vessel. Design pressure will therefore be nearer to operational pressure in the steam cushion that determines storage temperature and energy content. There are savings in the cylindrical part, the flat plate and the tangential as well as the axial cables.

4.4 Specific design features

Several of the more important design features of the reference design are discussed in the following paragraphs in some detail.

4.4.1 Cast iron blocks

Main dimensions of the cast iron blocks are specified in part (2) of Table IX. Blocks constituting the cylindrical part are solid, the ones of the flat plates are of hollow design.

The cylindrical part consists of 35 rings each ring being made up of six sectoral elements. Unit weight is 24.3 tonnes.

The flat plates, too, consist of sectoral elements. There are five outer rings, each consisting of six elements, an intermediate ring consisting of three elements, and a central block housing the pipe penetration. Mean block weight is 30.25 tonnes.

The cast iron blocks are machined at their faces. They are positioned by keys that are made of carbon steel St 50. In the flat plates, the keys also transmit transversal forces.

Fig. 32 shows design details of the flat plates featuring the step-arrangement that has been explained in para. 3.4.5.

4.4.2 Prestressing system

Data on the cables are found in part (3) of Table IX, the material specification in Table IV. Diameter of the tendons is 7.0 mm.

Axial prestressing requires 432 strands consisting of 72 wires each, or a total of 31104 wires. The strands end on both sides in header plates with holes that individually take up the wires the ends of which are plastically deformed outside the header plate. The header plate is designed for hydraulic prestressing (Fig. 33).

Tangential prestressing wires are shown in Fig. 34. They are packed tightly in the axial direction. Radially, there is a sheet of 1.5/2.0 mm thickness

between each layer, resulting in a radial pitch of $7.0 + 1.5 = 8.5$ mm. A bundle height of 140 mm therefore consists of about 17 layers.

The shoes supporting the tangential cables are shown in Fig. 35; data are given in part (3.5) of Table IX. In the cylindrical part, there are 36 shoes on the circumference, with an axial distance of 1.0 m. Each cast iron block of 2.0 m height supports $(2)(36/6) = 12$ shoes. In the flat plate, 72 shoes on the circumference are provided. A wire catcher is provided for safety reasons in case one of the outer wires fails.

Tangential cables will be installed with some degree of prestressing (about 500 N per tendon). Nevertheless, there will be an inhomogeneous radial stress distribution in the cable bundle. According to preliminary experimental investigations carried out by Siempelkamp, load variation will be of the order of $\pm 15\%$. Total cross-section of the tangential cables had, therefore, to be increased by a factor of $1/0.85 = 1.176$.

A further deviation from the ideal case is given by the shape of the tangential cables being not circular but rather polygonal. However, with the number of shoes being as large as $n = 36$ or 72, respectively, the difference turns out to be negligible.

For corrosion protection, Siempelkamp are considering the use of zinccoated cables. Care has to be taken to avoid hydrogen embrittlement during the pickling procedure preceding the coating process /78/. Axial cables may alternatively be protected by encapsulating the strands in oil-filled pipes.

4.4.3 Thermal insulation

Two rather different thermal insulation materials are required. Thermal insulation of the main body in between the shoes may be performed in a conventional way, e.g. by glass wool and an aluminum sheet cover. Thermal insulation between shoes and main body requires materials of low thermal conductivity but high compressive strength in order to minimize the area required and thereby the heat loss. (Ideally, the compressive strength should be the one of cast iron). Heat losses are thus proportional to $\lambda/\sigma_{\text{per}}$, where λ is the thermal conductivity and σ_{per} the permissible compressive strength of the thermal insulation material.

Data of several materials considered for pressurized thermal insulation are listed in Table XI. Originally, pure alumina (trade name RAPAL) had been considered. Although compressive strength of such materials is high, thermal conductivity is also very high and therefore $\lambda/\sigma_{\text{per}}$ is high.

Isographite (trade name SIGRAflex) was also considered. This material has rather interesting properties. Its transverse thermal conductivity (across the sheet) is lower by several orders of magnitude than the longitudinal thermal conductivity (along the sheet). On top of that, the transversal thermal conductivity decreases with pressure (the longitudinal one increases). Maximum allowable compression is not well known at present. If the value of cast iron, $656/3 = 219$ MPa, is assumed, $\lambda/\sigma_{\text{per}}$ may be extrapolated from existing data to be about 0.026.

Finally, however, "Klingerit 1000" was chosen for the reference design. Its data are given in Table XII. This material has been used for decades as a gaskets material at high pressures and temperatures; its properties are therefore relatively well known and do not require extrapolation. It is not expensive.

Its compressive strength is not very high but its thermal conductivity is rather low, resulting in a favourable $\lambda/\sigma_{\text{per}}$ -value.

It should be borne in mind that the compressive strength of Klingerit 1000 listed in Table XII applies only to layers of not more than a few millimeters. In order to obtain good thermal insulation, several layers have to be used (Table IX). They will be separated by thin (0.5 mm) sheets of stainless steel, possibly with a slightly roughened surface.

Heat losses of the reference PCIV have been calculated in Table XIII. The heat loss at design condition (fully charged) turns out to be about 400 kW of thermal power. Considering the high thermal capacity of the vessel content, this amounts to a loss of available energy of about 1 % in 24 hours.

4.4.4 Liner

The liner has to ensure leak tightness of the PCIV, in much the same way as in a PCPV. However, compression of the liner is larger in a PCIV than in a PCPV, because of the better material utilization of the PCIV. Two basic designs are possible:

(a) Strip liner

It is assumed that the castings as such are leak tight and only the block interfaces are to be sealed. This may be performed by sheet metal strips of rectangular cross-section that are positioned in the mould and welded after installation and prestressing of the whole PCIV.

(b) Integral liner

The internal surface of the PCIV is completely clad with sheet metal that is welded leak tight. The liner either has to be installed with some clearance in the unpressurized condition (floating liner, or more correctly: non-anchored liner, possible only in specific designs) or the liner may be anchored to the main body, e.g. by bolts that are screwed to the cast iron body and welded to the liner (anchored liner). A considerable number of bolts will be required. For the anchored liner, differential expansion between liner and main body due to temperature differences and prestressing has to be compensated by liner strain creating very high local stresses at the anchor points. Leak testing is extensive and costly. There is, however, no cumulation of differential expansion or contraction; consequently, there are no special problems at the penetrations. This system is therefore suitable for nuclear reactor pressure vessels with many penetrations in the cylindrical part and in the flat plates.

The PCIV as a TES vessel has very few penetrations and only at the poles. A floating liner that will get in contact with the wall of the main body by internal pressure and/or thermal expansion is feasible. However, an anchored liner of 1.0 mm stainless steel is chosen as preferred design for the Reference PCIV. The high ductility of stainless steel is an advantage, the large thermal expansion, i.e. $(17)10^{-6} \text{ K}^{-1}$ as compared to $(11)10^{-6} \text{ K}^{-1}$ for cast iron (Table III), a disadvantage.

Sheet metal will be installed into the PCIV, the rims will be welded together by the roll-bond process. Fastening to the cast iron main body will preferably be by local spot welding, using explosion-type spot-welding guns. A consi-

derable number of spot welds will be required but welding will be fast and relatively inexpensive. No metallurgical problems in the weld effected zones are envisaged.

In the transition zone between cylindrical part and flat part and at the pipe penetrations, relatively heavy stainless steel rings will be provided.

Future advanced designs may include the following alternatives:

- liner made of heat-resistant rubber, in particular silicone rubber that is produced even today for operating temperatures up to 400°C;
- no liner at all, but leak-tight castings and sealing of possible leaks (or metal spraying of the whole internal surface) and sealing of block interfaces by means of Klingerit, soft iron, or equivalent, or without gasket and sealing by prestressing only (as in modern turbine practice).

4.4.5 Internals

Use of the PCIV for TES according to the expansion principle does not require any substantial internals (only a spray pipe). In a displacement TES vessel, only strainers at inlet and outlet for equalizing flow patterns and avoiding undue turbulences and mixing are required. TES with internal steam generation, however, do require internals to achieve an orderly and sufficiently strong recirculation during the charge and discharge procedure; temperature stratification and loss of capacity during charging and uncontrolled boiling with incomplete steam/water separation during discharging are avoided in this way /51, 117, 118/.

The internals of accumulators with internal steam generation consist mainly of steam nozzles and a cylindrical shroud separating the riser and the downcomer section of the vessel cross-section. In addition, frustroconical shrouds are positioned in the upper part of the vessel (in the region of water level fluctuations) in order to stimulate controlled recirculation at an early date by capturing rising steam bubbles, leading them to the riser part and initiating natural recirculation (Figs. 30 and 9).

4.4.6 Pipe penetrations

Apart from the main penetrations for charge and discharge pipe or pipes, there may be several smaller penetrations for a safety valve, pressure transducers, drain pipes etc. Preferably, these should be combined with a charge or discharge pipe penetration. Coaxial pipe penetrations may be utilized in order to further reduce the number of penetrations.

Penetrating pipes are connected to the cast iron body by means of an internally situated flange that also takes care of possible liner movements. Large pipes may be provided with a Venturi shaped insert to limit flow rate in case of pipe failure.

4.4.7 General arrangement

Fig. 29 not only shows the reference vessel but also the general arrangement. The PCIV is supported via bearings on the foundation plate. One bearing is fixed, the others are either roller-bearings or friction-bearings employing teflon-coated surfaces.

Depending on climatic and other conditions there will or will not be an outer cover structure (left or right side of Fig. 29) that will be of light-weight construction. Its weight will be supported by the PCIV. Also wind pressure will be taken up by the heavy PCIV via radial struts.

For maintenance and inspection, a circular crane with cabin is provided avoiding the need for an elaborate stair and gallery system. Upper and lower flat plate are equipped with a manhole enabling internal inspection and maintenance.

Different arrangements may be suitable for a number of PCIVs being installed simultaneously or consecutively at the same site.

4.4.8 Installation and start-up

A possible installation, commissioning and start-up sequence may be as follows:

- site clearance, excavation
- casting of foundation
- temporary structure for installing lower flat plate
- installation of lower flat plate, starting at the center, proceeding outward
- installation of tangential cables on lower flat plate with slight stressing (about 500 N per cable) and subsequently applying part of prestressing hydraulically
- on an installation rig near the PCIV: assembling of cylindrical rings, installation of tangential cables with light stressing and subsequent applying part of prestressing hydraulically
- putting in place each cylindrical ring by means of a temporary crane (capacity: 150 tonnes)
- step by step with installation of rings: installation of liner
- after last cylindrical ring has been installed: erection of temporary structure for installing upper flat plate
- installation of upper plate, starting at the center, proceeding outward
- installation of tangential cables on upper flat plate with light stressing and subsequently applying part of prestressing hydraulically
- lifting of upper flat plate hydraulically, removing temporary structure
- letting down upper flat plate and connecting with cylindrical part
- installing axial cables, prestressing partly
- simultaneously: finalizing liner installation
- outer cover, crantage
- final axial prestressing
- final tangential prestressing, starting from flat plates
- inspection; checking strains

- cold hydraulic pressure test (50 % of design pressure); checking strains
- hot (100°C) hydraulic pressure test (85 % of design pressure); checking strains
- finish thermal insulation
- hot hydraulic pressure test (operating temperatures, 110 % of design pressure); checking strains and temperatures at outer surface
- final inspection, commissioning, hand-over
- put in service.

There is an alternative installation method: It consists in hydraulically lifting the whole structure as installed and adding another ring from below. This method has the advantage that the upper flat plate is completely installed on a rig at ground level and the need for elaborate temporary structures is greatly reduced. Also the large temporary site crane is not required.

The installation sequence for the horizontal alternative design will be similar. There will be a girder-type crane.

4.5 Specific stress analysis, transient conditions

4.5.1 General

Temperature gradients in the vessel wall will occur during normal operation of a hot going vessel as follows:

(1) Varying pressure accumulator

During charging, the temperature of the water content increases, parts of the vessel wall adjacent to the water content will be heated up by convective heat transfer, parts of the vessel wall adjacent to the steam cushion by condensation. During discharge, the temperature of the water content decreases, the wall adjacent to the water content is cooled by boiling heat transfer; the wall adjacent to the steam cushion, in particular the upper flat plate, will be cooled by stagnant steam only - which will be slightly superheated, at least in its boundary layer - and will remain hotter than the content of the vessel for quite some time.

(2) Displacement type

The transition zone between upper (hotter) and lower (colder) part of the water content induces temperature discontinuity stresses even in the steady-state condition (or more correctly stated, there is no steady-state condition other than complete temperature equalization). During charging and discharging, additional thermal stresses are induced that are larger than those of a varying pressure accumulator.

(3) Expansion-type

Conditions are similar to the varying pressure accumulator, but temperature change will be lower by about one order of magnitude. There will be essentially no thermal gradient at all if saturated steam of the same temperature is used rather than self-generated steam to expel the water content of the vessel.

4.5.2 Thermal stresses in varying pressure type and expansion-type storage vessels

Because of finite thermal conductivity, heat capacity and thermal expansion, temperature profiles are generated and thermal stresses induced in the vessel wall during charging and discharging. For simple geometries like plates, cylinders, spheres, or shells, and simple temperature changes like shocks or ramps, analytical solutions for the resulting thermal stresses are possible and may be found in the literature /119, 120, 127/.

(a) Cylindrical part

Solutions given below are applicable to thin-walled cylinders but may be used for thick-walled cylinders as an approximation.

The following symbols are used:

$T(t)$ time dependent temperature of storage medium

t time

α heat transfer coefficient storage medium/vessel wall

λ thermal conductivity

ρ density

β coefficient of thermal expansion

c specific heat capacity

s wall thickness

$$t^+ = \frac{\lambda}{\rho c} \frac{1}{s^2} t \text{ dimensionless time (Fourier Number)} \quad (4.25)$$

$$\alpha^+ = \alpha s / \lambda \text{ dimensionless heat transfer (Biot Number)} \quad (4.26)$$

$b = dT/dt$ temperature rise

For the special case of an unlimited thermal ramp:

$$T(t) = T_0 \quad \text{for} \quad t \leq 0$$

$$T(t) = T_0 + bt \quad \text{for} \quad t > 0$$

Fig. 36 shows the time-dependent stresses on the inside of a thin-walled cylinder subject to an unlimited ramp. The upper plots have a logarithmic, and the lower one has a linear stress scale. Dimensionless stress is given by /119, 120, 121/:

$$\sigma^+ = \frac{\sigma(1 - \nu)}{\beta E b s^2} \frac{\lambda}{\rho c} \quad (4.27)$$

This stress approaches a value of $-1/3$, faster or slower, depending on the Biot Number α^+ .

For practical purposes it is important to know the time after which the "quasi steady-state condition" (defined as constant temperature gradient and constant thermal stress in the wall and change of mean temperature at a constant rate) is "almost" (e.g. by 99 %) reached. This value $t^+(99\%)$ may be taken from the upper right-hand picture of Fig. 36. This occurs at the time when $\sigma^+ = 1/3 = (0.01)(1/3) = 1/300$.

If the temperature change is slow as compared to $t^+(99\%)$, quasi steady state calculation of thermal stress is permissible [121]. In equation (4.27), $\alpha^+_{\max} = -1/3$; $b = dT/dt$, and

$$\sigma = -s^2(dT/dt) \frac{\beta E}{1-\nu} \frac{\rho c}{\lambda} \frac{1}{3} \quad (4.28)$$

In the work of Pich [121] further corrections are given for thick-walled cylinders and spheres.

For processes of medium speed, any temperature function may be simulated by a polygonal shape and subsequently by a multitude of unlimited ramps. Resulting stresses are obtained by superposing stresses for the individual ramps. For quick and short temperature changes, this method is not very exact.

Fast transient are better represented by a temperature shock. In Fig. 37 stress is plotted against time for a temperature shock

$$\begin{aligned} T(t) &= T_0 & t &\leq 0 \\ T(t) &= T_0 + \Delta T & t &> 0 \end{aligned}$$

Dimensionless stress is defined as

$$\sigma^+_r = \frac{\sigma(1-\nu)}{\beta E \Delta T} \quad (4.29)$$

The lower part of Fig. 37 shows the maximum value of $(\sigma^+_r)_{\max}$ as a function of α^+ . For $\alpha^+ > 50$, as a first approximation $\sigma^+_r = 1$, and

$$\sigma_r = - \frac{\beta E \Delta T}{1-\nu} \quad (4.30)$$

For $\alpha^+ < 0.1$, on the other hand,

$$(\sigma^+)_{\max} = -\alpha^+/3 \quad (4.31)$$

(b) Flat plate

Relations given in (a) apply to a body that may move freely according to its mean temperature whereas movements resulting from temperature gradients are suppressed, as for instance in a circular ring with a radial temperature gradient. A freely supported plate may compensate a linear axial temperature gradient by warping. Actually, the flat plate is not freely supported but rather to be considered as fixed, and secondly there are discontinuity stresses at the transition to the cylindrical part. Relations given in (a) may therefore serve as a first approximation.

The flat plate of the reference design is of the girder type design. Heat transfer through the plate takes place by radiation and convection in the cavities and by conduction in walls. A fictitious thermal conductivity or temperature conductivity of the hollow plate may be defined.

4.5.3 Thermal stresses in displacement storage vessels

Thermal stresses in hot going displacement vessels differ from those in vessels with internal steam generation (Fig. 38). In the partly loaded condition, the lower part of the vessel is filled with cold or relatively cold water. The upper hot water part floats upon the lower cold water part. There will be an intermediate mixing zone. The thickness of the mixing zone increases as time proceeds. It is determined by thermal conductivity of the water, wall thickness and thermal conductivity of the vessel wall, and (to some extent) by convection. After a sufficiently long period of time, temperatures of upper and lower part will equalize (unless heat loss due to incomplete thermal insulation suppresses this effect). The thickness of the mixing zone may only be limited by complete charging or discharging the vessel. Enlarging the mixing zone reduces thermal stresses but, at the same time, reduces storage capacity due to exergy losses by mixing.

The temperature difference between upper (T_u) and lower (T_l) part of the displacement storage vessel induces thermal stresses in the cylindrical part even outside the charging and discharging operation, due to differential thermal expansion of upper and lower part and discontinuity stresses in the mixing zone where maximum flexural stress σ_b occurs at the point of maximum axial thermal gradient. This may be written as /95/:

$$\sigma_b = \frac{1}{6} \beta E \Delta T \frac{\sqrt{R s}}{l} 0.7063 \frac{A(A+B) - C(C+D)}{(A+B)^2} \quad (4.32)$$

$$\Delta T = T_u - T_l$$

$$A = \sinh K; \quad B = \cosh K; \quad C = \sin K; \quad D = \cos K$$

$$K = 1.2854 \sqrt{R s}$$

where l is the length of temperature transition (mixing length).

In addition to this steady state flexural stress, there will be a temperature ramp during the charging and discharging operation with ramp height ΔT and transition time

$$\Delta t = t_d \sqrt{l/L} \quad (4.33)$$

with

t_d ... charge (discharge) time

L ... internal height of vessel

The stresses resulting from this temperature ramp (to be calculated according to para. 4.5.2) are, of course, to be superimposed on the steady-state thermal stresses.

The critical point during discharge, i.e. upward movement of the transition zone (Fig. 38), is the lower end of the mixing zone. At the inside of the vessel wall there will be tension due to temperature discontinuity and tension due to the rampwise temperature drop.

The charging operation is somewhat less critical. Tension will occur at the outside of the vessel at the upper end of the transition zone which is moving downward. The rampwise temperature increase induces compression at the inner surface of the vessel which is, of course, less critical than the tension during discharge.

4.5.4 Permissible thermal stresses

Thermal discontinuity stresses are secondary stresses and, as such, limited according to the ASME Code /104, 105/ to $2 S_m$, and thermal peak stresses to $3 S_m$. All thermal stresses including local peak stresses are, however, to be considered for fatigue analysis. Permissible alternating stress depends on the number of cycles that will occur during the lifetime of the vessel. A fatigue curve is required on at least the permissible alternating stress for 10^4 to 10^5 cycles. In the ASME Code /104, 105/ fatigue curves are available for several materials but not for cast iron. Other sources are not known. There is only data on the endurance limit (Dauerwechselfestigkeit) for 10^7 cycles, according to the work of Hempel /103/. These data are being used for fatigue analysis. They are pessimistic in that the fatigue limit will certainly be higher for 10^4 to 10^5 cycles than for 10^7 cycles. According to Siempelkamp, the difference should be substantial. As mentioned in para. 3.2.1, fatigue tests for 10^4 to 10^5 cycles are being carried out by Siempelkamp but results will only begin to become available during 1977.

The following calculation is therefore based on the conservative values of Hempel /103/. To eliminate the influence of the mean stress, the simplification was made that stress amplitude must not enter the tensile range. This means that the range is two times mean stress (amplitude is equal to mean stress). As can be seen from Fig. 14, this simplification seems to be reasonable if mean stress is in the range of 100 to 200 MPa as in our case.

The above requirement of zero tensile stress for cyclic load (thermal stresses during discharge of the hot going vessel) is tantamount to the requirement of no gapping at the (inner) surface of the cast iron wall. Such "surface gapping" must not be confused with the gapping of the whole cross-section as described in para. 3.4.3. Surface gapping due to thermally induced tension does not mean a sudden and large opening up of the whole cross-section but rather a relatively slow and limited opening up of the innermost part of the vessel wall only. The width of the surface gaps will for a plain surface depend on block size (tangential and axial length of blocks); it may be greatly reduced by structuring the block surface by means of fins, or of crevices, in both the axial and radial direction. In this way the gap width is split up into many small deformations and actual surface gapping at the block interfaces will be very

small or even suppressed despite considerable tensile stresses. Furthermore, temperature shocks, as such, may be softened by design measures which, however, point already into the direction of a cold going vessel.

Fig. 39 shows stress distribution in the PCIV at full pressure and temperature. In the cylindrical part, compressive stress in the cast iron body is

$$\sigma = f_{cc}(1 - 1/F_{gh}) \quad (4.34)$$

For the reference design:

$$f_{cc} = 164 \text{ N/mm}^2 \text{ (at } 275^\circ\text{C, see Table II)}$$

$$F_{gh} = 1.5$$

$$\sigma = 164 (1 - 1/1.5) = 54.7 \text{ N/mm}^2$$

Permissible tensile stress due to thermal transients must not exceed 54.7 N/mm^2 .

Conditions are somewhat different in the flat plate. Compressive stress at the inside is

$$\sigma_i = f_{cc}(0.5 + 0.5/F_{gh}) \quad (4.35)$$

and at the outside:

$$\sigma_o = f_{cc}(0.5 - 0.5/F_{gh}) \quad (4.36)$$

For the reference design:

$$f_{cc} = 85 \text{ N/mm}^2 \text{ (at } 275^\circ\text{C, see Table II)}$$

$$F_{gh} = 1.5$$

Compressive stress at the inside for the hot, pressurized condition will be

$$\sigma_i = 85(0.5 + 0.5/1.5) = 70.8 \text{ N/mm}^2$$

and at the outside

$$\sigma_o = 85(0.5 - 0.5/1.5) = 14.2 \text{ N/mm}^2$$

Tensile stresses due to thermal transients must not exceed these values if tensile stress is to be avoided.

The following cases have to be investigated:

- (a) A temperature drop of the vessel content (discharge) causes tensile thermal stress on the inside, a reduction of compression or even tension (gapping) and, furthermore, discontinuities between cylindrical part and flat plate. Tensile thermal stress must not exceed 54.7 N/mm^2 in the cylindrical part and 70.8 N/mm^2 in the flat plate.
- (b) A temperature increase of the vessel content (charging) may lead to a reduction of compression at the outside of the vessel which is less critical. Discontinuities are, however, to be considered.

4.5.5 Permissible temperature variations in varying pressure type and expansion-type storage vessels

As mentioned in para. 4.5.2, the investigation of temperature ramps is of fundamental importance since any temperature variation may be split up accurately in a number of temperature ramps and thermal stresses resulting from the ramps may be superimposed. Permissible limited temperature ramps (defined by temperature change ΔT and transition time Δt) will be calculated and plotted in a ΔT vs. Δt diagram (Beckmann plot /122/ Fig. 40). The special case of permissible temperature shock ($\Delta t = 0$) is given by the ordinate, and the permissible unlimited temperature ramp ($\Delta t = \infty$) as an asymptote.

(a) Cylindrical part

Permissible thermal stress is 54.7 N/mm^2 (para. 4.5.4). Important data for heat transfer, heat conduction, and thermal storage are:

$$\begin{aligned}
 \alpha &= 10\,000 && \text{W/m}^2\text{K (for discharge; less for charging)} \\
 s &= s_c = 0.3 \text{ m} && \text{(approx.)} \\
 \lambda &= 50 && \text{W/mK} \\
 \rho &= 7200 && \text{kg/m}^3 \\
 \beta &= (12)10^{-6} && \text{K}^{-1} \\
 c &= 0.55 && \text{kJ/kg K} \\
 E &= 86250 && \text{N/mm}^2 \\
 \nu &= 0.28
 \end{aligned}$$

$$a = \lambda / \rho c = \frac{50}{(7200)(550)} = (12.6)10^{-6} \text{ m}^2/\text{s} \quad (4.37)$$

(a1) Permissible temperature shock

From equation (4.27) the Biot Number $\alpha^+ = 60$ is obtained. After $t^+(99\%) = 2$ or $t(99\%) = 14256 \text{ seconds} = 3.96 \text{ hours}$, 99 % of the quasi steady state stress is reached. The lower part of Fig. 37 shows the permissible temperature shock. The Biot Number $\alpha^+ = 60$ yields $\sigma^+ = 1$, and according to equation (4.30)

$$\underline{\underline{\Delta T}} = \frac{54.7 (1 - 0.28)}{(12)10^{-6} (86250)} = \underline{\underline{38 \text{ K}}}$$

(a2) Permissible temperature gradient

Permissible unlimited temperature gradient for decreasing temperature is calculated by means of equation (4.28):

$$\sigma = (54.7)10^6 = 0.3^2(dT/dt) \frac{(12)10^{-6}(86\ 250)10^6}{(1 - 0.28)} \frac{(7200)(550)}{50} \frac{1}{3}$$

$$\underline{\underline{dT/dt}} = 0.016\text{ K/s} = 0.96\text{ K/min} = \underline{\underline{57.5\text{ K/h}}}$$

Actually more important than the two special cases shock and unlimited ramp is the limited ramp. Using the principle of superposition and equation (4.27), letting $\sigma = \sigma_{\text{per}}$, and using values from Fig. 36, permissible temperature ramps were elaborated (Fig. 40). For a start from cold to hot condition ($\Delta T = 275\text{ K}$), about 5 hours are required if only the cylindrical part is considered.

(b) Flat plate

Permissible stress in the flat plate is higher (70.8 N/mm^2) but wall thickness is higher too. Permissible temperature gradients are smaller than for the cylindrical part.

When calculating thermal conduction and storage withing the flat plates, the hollow structure of the girder type flat plate (Table VI) is to be accounted for. There are two intermediate horizontal walls between outer and inner flange. The interspaces are partitioned into a multitude of cells of 1 m by 1 m plan view (Fig. 41). Effective thermal conductivity λ_{eff} of the hollow flat plate is calculated as follows:

$$\text{Upper and lower flanges:} \quad s_1/\lambda_1 = 0.32/50 = 0.0064\text{ m}^2\text{K/W}$$

$$\text{Horizontal intermediate walls:} \quad s_2/\lambda_2 = 0.08/50 = 0.0016\text{ m}^2\text{K/W}$$

$$\text{Cells:} \quad \lambda_3/s_3 = (50)(2)(0.06)/0.92 + (18)(0.88) = 22.36$$

and therefore

$$s_3/\lambda_3 = 0.047\text{ m}^2\text{K/W}$$

using a heat transfer coefficient of air $\alpha = \alpha_r + \alpha_c = 18\text{ W/m}^2\text{K}$, with

$$\alpha_r = 16\text{ W/m}^2\text{K} \text{ by radiation, and}$$

$$\alpha_c = 2 \text{ " by convection}$$

$$h/\lambda_{\text{eff}} = \Sigma s_i/\lambda_i = 2 s_1/\lambda_1 + 2 s_2/\lambda_2 + 3 s_3/\lambda_3 = 0.15\text{ m}^2\text{K/W}$$

$$\underline{\underline{\lambda_{\text{eff}}}} = h/0.15 = 3.56/0.15 = 23.7 \div \underline{\underline{24\text{ W/m}^2\text{K}}}$$

that is about half of the thermal conductivity of cast iron.

The equivalent density ρ_{eff} of the hollow structure is now calculated. The cast iron volume for 1 m² of flat plate, i.e. a volume of $V = 3.56$ m³, is:

$$V_{ci} = (2)(0.32) + (2)(0.08) + (3)(0.06)(2) = 1.16 \text{ m}^3$$

and the corresponding cast iron mass: $G_{ci} = \rho V_{ci} = (7200)(1.16) = 8352$ kg. Therefore:

$$\rho_{eff} \approx G_{ci}/V = 8352/3.56 = \underline{\underline{2400 \text{ kg/m}^3}}.$$

The effective temperature conductance a_{eff} in the flat plate becomes:

$$\underline{\underline{a_{eff}}} = \frac{\lambda_{eff}}{\rho_{eff} c} = \frac{24}{(2400)(550)} = \underline{\underline{(18.2)10^{-6} \text{ m}^2/\text{s}}}$$

that is more, by 44 %, of the temperature conductance of solid cast iron which is $(12.6)10^{-6} \text{ m}^2/\text{s}$.

(b1) Permissible temperature shock in the flat plate

Biot Number becomes

$$\alpha^+ = \alpha h/\lambda_{eff} = (10\ 000)(3.56)/24 = 1483$$

From Fig. 33, $t^+(99\%) = 1.7$ is obtained and using equation (4.25), $t(99) = (1.18)10^6$ seconds or 330 hours for reaching steady state conditions during a temperature ramp. Any temperature change that takes place in much less than 330 hours is felt by the flat plate rather as a temperature shock. With $\alpha^+ = 1483 \approx \infty$, $\sigma_r^+ = 1$ is obtained from Fig. 37 and the permissible temperature shock becomes

$$\underline{\underline{\Delta T}} = \frac{(70.8)(1 - 0.28)}{(12)10^{-6}(86\ 250)} = \underline{\underline{49.3 \text{ K}}}$$

This is more than for the cylindrical part.

(b2) Permissible temperature gradient

The permissible unlimited temperature ramp (gradient) is obtained from equation (4.28):

$$\sigma_{per} = (70.8)10^6 = 3.56^2(dT/dt) \frac{(12)10^{-6}(86\ 250)10^6}{(1 - 0.28)} \frac{(2400)(550)}{24} \frac{1}{3}$$

$$\underline{\underline{dT/dt}} = 0.0002 \text{ K/s} = 0.0127 \text{ K/min} = \underline{\underline{0.764 \text{ K/hr.}}}$$

that is about two orders of magnitude less than for the cylindrical part. Fig. 40 shows permissible temperature ramps. The flat plate requires about 380 hours for start-up ($\Delta T = 275$ K). For $\Delta t = 0.5$ hours, permissible temperature changes are about equal for flat plate and cylinder.

4.5.6 Permissible temperature variations in displacement storage vessels

The critical item in the calculation of thermal stresses in displacement storage vessels is the determination of the mixing length or, exactly, the determination of temperature as a function of vessel height and time. No theoretical model exists as yet. If there was only heat conduction, analytical solution of the problem would be possible by Fourier analysis. Experimental investigations (for instance, at the displacement storage vessels of the Sendling district heating plant of the Munich, FRG, municipality /95/) have, however, shown a considerable effect of the water velocity during loading and discharging. Data were as follows:

Vessel dimensions:	$L = 30$ m, $R = 1.89$ m, $s_c = 0.015$
Upper/lower temperature:	$160/60$ °C
Temperature difference:	100 K
Thickness of mixing zone (80 to 140°C):	
at a water velocity of 6.15 mm/s:	$l = 1.1$ m
" " " " " 8.75 mm/s:	$l = 3.3$ m

This measured dependence of the mixing zone on water velocity (which is proportional to discharge velocity) leads to the somewhat paradoxical result that thermal stresses are lower for faster discharge (of course, at increased mixing losses). Dimensionless height of the mixing zone was $1.1/30 = 0.04$ (4 %) and $3.3/30 = 0.11$ (11 %) respectively. For the following investigations, the mean value (7.5 %) is used.

Thermal stresses of the reference design used as a displacement storage vessel are now calculated.

Vessel dimensions:	$L = 71$ m, $R = 6$ m, $s_c = 0.3$ m (approx.)
Upper/lower temperature:	$275/105^\circ\text{C}$
Temperature differences:	170 K
Mixing zone:	$l = (0.075)(71) = 5.33$ m

From paragraph 4.5.3:

$$K = (1.2854)(5.33) / \sqrt{(6)(0.3)} = 5.11$$

$$A = \sinh 5.11 = 83$$

$$B = \cosh 5.11 = 83$$

$$C = \sin 5.11 = 0.92$$

$$D = \cos 5.11 = 0.39$$

and from equation (4.32):

$$\sigma_b = \pm (12)10^{-6}(86\ 250)(170) \frac{\sqrt{6}(0.3)}{5.33} 0.7063 \frac{(83)(83 + 83) - 0.92(0.92 + 0.39)}{(83 + 83)^2}$$

$$\sigma_b = 15.7 \text{ N/mm}^2$$

=====

This is not very much. However, thermal stress during discharge due to the temperature ramp given by the mixing zone is to be added. This additional ramp thermal stress shall not exceed:

$$\sigma_{\text{ramp}} = 54.7 - 15.7 = 39 \text{ N/mm}^2$$

The equivalent temperature change is therefore

$$\Delta T_{\text{eq}} = \Delta T \sigma_{\text{per}} / \sigma_{\text{ramp}} = (170)(54.7) / 39 = 238 \text{ K}$$

From Fig. 40, minimum transition time is

$$\Delta t = 4.0 \text{ hours}$$

and therefore the minimum discharge time is

$$t_d = \Delta t L / l = (4.0)(71) / 5.33 = 53.3 \text{ hours} = 2.2 \text{ days}$$

which is outside the range of interest for high pressure TES.

The flat plates are even more critical than the cylindrical walls. The steep temperature gradient of the transition zone will have to be kept away from the flat plates at all cost. In other words, the upper flat plate would always have to be kept near the upper temperature T_u , the lower flat plate near the lower temperature T_l . This will mean some sacrifice in storage capacity and it will mean operational restrictions such as automatic stopping the charge or discharge process when the transition zone approaches the plates; this may be signalized by means of thermocouples. Even so, it is doubtful whether thermal stresses in the flat plate can be kept sufficiently low without inside thermal insulation, considering the thermal conduction processes during the intermission between the charging and discharging operation.

4.5.7 Possibilities of improvement

It has been shown that the flat plate is the most critical item in hot going vessels for varying pressure as well as for displacement storage (where, however, thermal stresses even in the cylindrical part are prohibitive). There are possibilities for reducing thermal stresses in the flat plate by increasing the effective thermal conductivity of the flat plate ($\lambda_{\text{eff}} = 24 \text{ W/m}^2\text{K}$ for the reference design). This may be achieved by one or more of the following measures:

(a) Reducing the number of horizontal intermediate walls from two to one:

$$h / \lambda_{\text{eff}} = 0.0128 + 0.0016 + (2)(0.0498) = 0.114$$

$$\lambda_{\text{eff}} = 3.56/0.114 = 31.2 \text{ W/mK}$$

(only a slight improvement),

(b) Avoiding horizontal intermediate walls altogether:

$$h/\lambda_{\text{eff}} = 0.0128 + 0.0559 = 0.0687$$

$$\lambda_{\text{eff}} = 3.56/0.0687 = 51.8 \text{ W/mK}$$

- (c) Perforating the intermediate walls (increased convection and radiation); effect similar to but less than (a) or (b).
- (d) Partly filling interspace with a suitable high-boiling fluid,
- (e) Utilizing closed heat pipes for heat transmission between outer and inner flange,
- (f) Utilizing closed heat pipes for heat transmission between outer flange and storage content.

With (d), (e) or (f), almost any improvement of λ_{eff} seems possible, but with a more complicated design and with considerable additional cost.

4.5.8 Conclusions from thermal stress investigations

Results of and conclusions from the investigation of thermal stresses and strains, using conservative data, may be summarized as follows.

- Temperature shocks are permissible in the cylindrical part up to 38 K, in the flat plate up to 49 K.
- Larger temperature changes require time, i.e. a temperature gradient. Unlimited gradients are permissible up to 60 K/h for the cylindrical part and up to about 1 K/h for the flat plate (only with somewhat exotic designs of the plate are faster transients permissible). A limited temperature gradient is permissible for the plate up to 60 K in one hour whereas 120 K requires 120 hours. Although slow start-up and shut-down (requiring 360 hours = 15 days for 275 K) may be acceptable, practical operational temperature changes are limited to 60 K.
- The basic possibilities of pressurized hot water storage do require about the following operational temperature changes:

Varying pressure type, 60/6 bar:	275 - 160 = 115 K
Displacement type, 60 bar:	275 - 105 = 170 K
Expansion type, 60/42 bar:	275 - 255 = 20 K
Expansion type, with recharge, 60/55 bar:	275 - 270 = 5 K
- The use of the hot going PCIV for displacement storage vessel is severely restricted by the large temperature change, the short transition time of the mixing zone passing-by, and the additional thermal discontinuity stresses. Necessary discharge times of more than two days render this concept unfeasible.

Use of the cold going PCIV will be required for displacement storage vessels. It should, however, be noted that all the figures mentioned are rather conservative and preliminary, since the results of the fatigue tests which are being carried out by Siempelkamp are not yet available.

- Another and probably more severe obstacle against using hot going PCIV for displacement type storage is the low safety factor in cold conditions which prevents the application of the advanced design base for self-stressing vessels. Contrary to expansion-type and varying pressure type vessels, temperature and pressure of the displacement storage vessel are not coupled. When partly charged, pressure will be determined by the temperature of the upper hotter part (charge condition) whereas temperature in the lower colder part will correspond to discharge condition.
- The varying pressure accumulator (with internal steam generation) may be used with the hot going PCIV if the temperature and thus pressure change is severely restricted (for instance to 250/190°C, 40/12 bar, for 1 hour discharge time or to 250/200°C, 40/16 bar for 30 minutes discharge time), which means sacrificing storage capacity and reducing the advantage of the PCIV. Combined internal steam generation plus expansion may be interesting in special cases for stop-gap duty.
- Expansion type steam and/or feed water storage, either with or without steam recharge, poses no problems for the use of the hot going PCIV, even at very fast discharges.

4.6 Specific Code problems

General Code problems of the PCIV have already been dealt with in paragraph 3.6. For the hot going PCIV (with cold going cables) as envisaged for TES applications, further points are to be considered:

- Self-stressing of the hot-going PCIV with advanced design base means that full internal pressure will be permissible only at elevated temperature. This may necessitate special safety devices that open up at a lower than design pressure if the PCIV is in the cold condition. In a way, this requirement is similar to the requirement for Light Water Reactors where the cold pressurized condition is to be avoided because of brittle fracture considerations.
- For the same reason, cold hydraulic pressure testing will only be possible at reduced pressure whereas the hydraulic test at full (or 1.1 times) design pressure requires operating temperature.
- Presently, there is an urgent need of a fatigue curve of cast iron for the number of cycles and temperatures of interest.
- Inspection of the hot going PCIV before start-up and during the lifetime is easier than inspection of the PCPV and the cold going PCIV. Any failures of liner or thermal insulation will be easily located and repaired.

5. CONSEQUENCES OF USING PCIV FOR PRESSURIZED HOT WATER STORAGE

The switch from the welded steel pressure vessel to the PCIV for TES duties and thus to higher pressures and much higher unit sizes and small numbers of units entails a series of consequences regarding steam conditions, flow sheet, etc. which will be discussed in the following paragraphs.

5.1 Unit size limitations

For welded steel vessels, optimum design pressure is about 2 MPa (20 bar; 300 psi /31/). With this design pressure, a very large vessel can technically be built; however, there is an economic limit /31, 37/ given by

- maximum diameter of vessels that may be transported in one piece (or two pieces); and
- maximum diameter given by the maximum wall thickness that does not, according to the ASME Code, require annealing of the whole shell after welding.

Both conditions practically limit the vessel diameter to about 4.75 m (15 ft.). With an L/D ratio of 8.0, length (height) of the vessel is 38 m and volume about 600 m³.

For the PCIV, there is no diameter or length (height) limit due to transportation or welding problems. Size, shape and weight of the individual blocks (castings) will be chosen so as to satisfy foundry requirements and to facilitate machining as well as handling and transportation to the site. Weight of the blocks will, therefore, be limited to perhaps 20 or 30 tonnes, as the case may be. As will be shown later (paragraph 6.2.3) there is, therefore, up to a certain limit, a marked decrease of specific PCIV cost with size, due to the reduction of surface area, thermal insulation etc.

There may, however, be limitations of the vessels size that are specific for the individual site:

- The height of the PCIV may be limited (for instance, due to a nearby airport),
- or subsoil strength may, above a certain size of the vessel, be insufficient to cope with the weight of the vertical PCIV without a large, heavy and expensive foundation plate.

Under normal conditions, there should be no problems of accomodating a PCIV of the size and weight of the reference vessel (internal diameter: $D = 12$ m, internal height: $L = 71$ m, overall height: $L_0 = 78$ m, internal volume: $V = 8000$ m³, total weight, empty: $G_0 = 10\,720$ tonnes, total weight, fully charged: $G = 16\,500$ tonnes). Even somewhat larger vessels may be feasible and economic. Very much larger vessels will, however, constitute no savings. Practical limit sizes may be roughly as follows:

for $p_0 = 10$ MPa (100 bar) pressure:

$$D = 12 \text{ m}, L = 88 \text{ m}, V = 10\,000 \text{ m}^3, pV = 1\,000\,000 \text{ bar m}^3$$

for $p_0 = 3$ MPa (30 bar) pressure:

$$D = 20 \text{ m}, L = 88 \text{ m}, V = 15\,500 \text{ m}^3, pV = 465\,000 \text{ bar m}^3.$$

5.2 Steam conditions

The higher design pressures of the PCIV steam storage system also mean higher working steam pressures. This in turn means higher enthalpy drop during expansion, lower mass flow rates, much lower volumetric flow rates, and a cost reduction for the following components of the peak power plant (regardless of flow sheet):

- piping system, valves;
- turbine and condenser;
- cooling tower and cooling water handling system.

In addition, higher working steam pressure will facilitate the use of the main (base load) turbine set for peaking duty by means of the TES system.

5.3 Flow schemes

5.3.1 General

Which flow sheet is suitable for a steam power plant with integrated TES depends on many considerations, for instance:

- on the kind of primary energy source used which determines whether the primary energy source is available at all times (nuclear, coal) or not (solar) and determines to some extent the pressure and temperature of live and reheat steam and the size of the plant;
- on whether the main (base load) turbine set may be overloaded for peaking or not (necessitating a separate peak load turbine per one or more main sets);
- as to the purpose of power generation from the TES system (peak load generation, load following, instantaneous or stop-gap reserve, spinning reserve);
- as to the type of storage system (varying pressure type, expansion-type or displacement-type);
- as to whether by the TES plant steam or hot water is provided or both (steam or feed water or combined feed water and steam storage);
- as to the type of the storage vessel (welded steel, PCIV, underground) and thus the pressure of the TES working steam and the number of accumulators;
- as to whether or not superheat (e.g. by means of high-boiling liquid storage) is provided to an extent to bring the temperature of the TES working steam anywhere near the temperature of the live steam.

There is thus a large number of possible combinations of criteria. It should, however, be noted that not all combinations are practical. The following discussion of flow sheets starts with the ones based on a separate peaking turbine; these are at the same time typical for peak load generation, for steam storage, and for large nuclear sets the generator of which is usually of limit size and may not be forced for peaking (exception: Heavy Water Reactors whose turbine/generator sets presently are built with smaller than limit size) and where any change in the main plant to accommodate TES (except a few small steam or feed water lines for charging the TES plant and the corresponding return lines) would be difficult. Flow sheets in which the main turbine/generator set may be overloaded for peaking purposes using TES steam are discussed afterwards; they are typical for feed water or combined feed water and steam storage and for coal-fired and solar steam power plants.

5.3.2 Varying pressure accumulators

Fig. 42 shows a flow sheet for a nuclear plant with TES and separate peaking turbine. The flow sheet is suitable for welded steel vessels in that it features sequential discharge of groups of vessels, a method that requires a substantial number of vessels /37/.

A Pressurized Water Reactor (PWR) is shown as the heat source; with only minor modifications, the flow sheet applies, however, to other reactor types, too. The main accumulators are of the varying pressure type. This type of steam storage features the following advantages: possibility of quick start-up, internal steam generation, absence of recirculation pumps, and almost complete use of the heat stored in the steel of the accumulator walls. The main accumulator vessels are charged and discharged stepwise and in groups. There are three steam lines of different pressures; the steam raised in the accumulators is throttled to the next lower steam line pressure.

Peak load is carried by a separate peaking turbine and not by forcing the main base load set. The peak load turbine is of a simple robust design which can accommodate rapid thermal expansion during frequent starts and stops. It features separate double-flow casings for high, medium and low pressure steam, but there are neither steam bleeds nor intermediate dryers of any kind.

In order to keep exhaust wetness low and internal efficiency high, the high medium steam is superheated by means of superheaters and a number of superheat accumulators of the displacement type, charged by live steam. The degree of superheat chosen limits exhaust wetness to 11 % in the three casings. Fig. 43 shows the enthalpy-entropy-diagram of the expansion in the peak load and in the main turbine.

Fig. 44 shows another arrangement with varying pressure type accumulator vessels and superheater, particularly suitable for instantaneous reserve or stop-gap plants /42/. There is considerable steam throttling involved, storage efficiency therefore is moderate which is, however, not all that important for short load durations. The flow sheet is very simple which is an advantage from the operational point of view. From the main storage vessel steam is taken off, throttled and superheated by means of high pressure hot water discharged from a separate superheat vessel. After having been cooled in the superheater the water is throttled and introduced into the main accumulator vessel via the charge pipe. There is only a moderate decrease of pressure and temperature in the superheater storage vessel, insofar as water volume is replaced by steam produced from the hot water. Because of the expansion effect of the steam volume - which saves a recirculation pump - this type of heat storage has been called expansion-type storage. Having no recirculation pumps, the storage system will start up promptly and will immediately supply superheated steam. Steam flow rates as well as the degree of superheat may be controlled with simple control valves.

The design principle of the internals of the main storage vessel is similar to the one of the Ruths-type steam storage vessels of the Charlottenburg plant /61, 63, 64, 74/. However, the flow direction was reversed by arranging the flow guiding frustracones in the reverse orientation. The central channel is now acting as downcomer, the annular channel as riser. Heat stored in the cast iron blocks and liberated during discharge now stimulates the water circulation rather than retarding it, and better use is made of the steam space with its central steam nozzle /23/.

5.3.3 External steam generation with displacement type accumulators

Fig. 45 shows an alternative flow sheet of a steam storage plant /37/. The welded steel accumulators are of the displacement type, with external steam generators. A High Temperature Reactor is shown as the heat source. Contrary to earlier similar proposals /33...36/, there are no recirculation pumps for either main or superheat accumulators. Hot water is taken from the superheat accumulators, cooled in the superheater (where it superheats the high and medium pressure steam of the peaking turbine), and subsequently its pressure energy serves for recirculating the content of the main (displacement) accumulators by means of a jet pump. There is no need for stepwise and sequential operation, so there are operational advantages. On the other hand, first cost may be slightly higher due to the cost of the steam generator units.

5.3.4 External steam generation with expansion-type accumulators

The following Figures show flow sheets that have been developed for the peculiarities of steam storage by means of PCIV: few large accumulator vessels and high design pressures, as mentioned above /42/.

Fig. 46 shows a flow sheet in which superheat and main accumulators are designed as expansion-type accumulators. The arrangement permits high storage efficiencies and, compared to displacement storage with recirculation, there are no recirculation pumps in operation during peak load operation. This flow sheet will therefore be especially suitable for covering daily load peaks by excess energy from nuclear or coal power plants. Steam of three different pressure levels is generated outside the expansion-type accumulator in a simple manner by means of flash tanks. For the high and the medium pressure steam, superheating is provided by means of a high pressure superheat accumulator of the expansion type.

There are three turbine casings in parallel. Control valves at the inlet of the flash tanks and control of the water flow rate from the superheat accumulator are used to achieve constant steam conditions at turbine inlet, constant flow rates and constant output over the whole discharge period. Water from the last flash tank is collected in a water tank at a pressure of slightly above atmospheric and corresponding temperature. During charging of the main accumulators, the water is pumped into the main accumulator and is, at the same time, heated directly or indirectly by means of bleed steam. It should be noted that the water tank requires only about half the volume of the main accumulator, because a substantial portion of the main accumulator content is evaporated at the high pressures of a PCIV and, in addition, there is a substantial difference in density.

Fig. 47 shows a variant of the plant with expansion-type accumulators according to Fig. 46. The separate superheat accumulator has been omitted. Superheating the high and medium pressure steam for the peak load turbine is achieved by means of branched-off high pressure water from the main accumulator. The saturation temperature of the main accumulator is therefore an upper limit for the steam temperature. For this reason, the flow sheet is somewhat changed in that high pressure steam is not expanded all the way to condensation but only to medium pressure; in a common medium pressure superheater it is reheated and steam from the medium pressure flash tank is superheated.

Thermal and flow data for the flow sheet according to Fig. 47 are given in Fig. 48. Data are based on constant power output and constant steam data and flow rates at the inlet to the three turbine casings.

5.3.5 Expansion-type feed water and steam storage

New flow sheets for fossil-fired, in particular coal-fired, stations are presented in Figs. 49, 50 and 51. They are based on combined steam and feed water storage in an expansion-type arrangement at a storage pressure suitable for PCIV. Charging is carried out by feed water and throttled live steam which are being mixed within the PCIV.

Two flow sheets are possible with the combined feed water storage system: Steam from the evaporator may be piped to the main turbine if it is designed for taking overload (Fig. 50 and Fig. 51) or to a separate peaking turbine arranged in parallel to the main set (Fig. 49). The peaking turbine is connected to the main set by two working steam lines: one from the cold reheat line supplying steam from the evaporator and possibly additional steam from the main set; the other one from the cross-over taking up steam that is in excess of the maximum steam flow rate of the LP-part of the main set. If it is required not to disturb the main set at all during peaking, the peaking turbine would have to use the steam from the evaporator and the steam from all the bleed nozzles that is not now required for feed water heating because hot feed water is supplied by the storage systems. Such an arrangement would, however, make the peaking turbine rather complicated.

Generally, overloading the main set has a series of implications. Closing all bleed nozzles theoretically would increase turbine power by about 15 %. At the same time, exhaust steam rate would increase by about 45 % /48, 12/. Related to the whole turbine set, this would correspond to an overdesign of about 30 %. In other words, for generating 1 kW of additional peak power, the main turbine would have to be larger by 2 kW, or, reversely, any existing overcapacity of the main set can be utilized only by one half with feed water storage. Besides, large saturated steam turbines such as used in connection with present-day nuclear reactors feature stage-drying for reducing exhaust wetness in order to limit blade erosion and turbine efficiency reduction due to steam wetness. A certain reduction of bleed flow may technically be possible, closing feed flow down completely certainly is not.

Overloading the main set does not therefore seem to be very attractive for new plants. It may be interesting for existing turbine sets with overcapacity available. A recent investigation /48/ has shown, however, that in the US the number of existing plants where Retrofitted Thermal Energy Storage (RTES) could be applied is rather limited.

In Figs. 49 and 50, there is a flash evaporator and a HP pump for the stored feed water; in Fig. 51, this is replaced by a HP mixing preheater acting as flash evaporator and a HP feed pump. Thermodynamically there is no great difference between these two solutions.

Fig. 52 shows a flow sheet of a solar power plant with PCIV varying pressure storage for instantaneous reserve (suitable for compensation during cloudiness), acting alternatively or simultaneously as feed water storage. From the live steam

line and from the cross-over line of the solar thermal electric power plant, there are branch connections to the storage plant which consists of a varying pressure type steam storage vessel and optionally a superheat/desuperheat storage, that is a tank containing a high-boiling liquid. Instead of the varying pressure type steam storage vessel, other types such as displacement type or expansion type vessels and flow sheets /42/ may be used. In the example, a mixing condenser and a dry cooling-tower are shown.

In Fig. 52, there are two additional features to take care of excess insolation to the steam generator as may be available at high noon, particularly during summer. (It is to be noted that only the steam generator, but not HP + LP turbine and condenser/cooling tower will be designed for peak insolation). The first feature is a feed water temperature control valve, mixing hot water from the storage tank and warm water from the feed water tank; in case of excess insolation only warm water is supplied to the steam generator. This feature may be considered as feed water storage in the sense that discharge is not actuated by a demand peak but rather by a supply deficit (for instance, during the early morning or late afternoon, or in case of clouds passing by), whereas during an insolation peak the full capacity of the steam generator is made use of. The second feature is a live steam by-pass line supplying excess live steam directly to the storage plant /46/.

A different arrangement, with expansion-type storage in a combined feed water arrangement for a solar power plant, is shown in Fig. 53. Peaking capacity is higher than in the arrangement as per Fig. 52 because of the series of flash evaporators.

5.4 Storage efficiency

Table II shows that storage type and flow sheet have a bearing on the various losses of available energy (exergy) and thus on storage efficiency. Values of storage efficiency given in Table II are for peaking operation, that is for a large number of annual discharges. For very low numbers of annual discharges, i.e. for pure stand-by or reserve duty, storage efficiencies may be much lower due to increased effects of heat losses; however, for these cases, storage efficiency is not of particular interest anyway.

The relatively high storage efficiencies of the feed water storage systems are somewhat academic in that they do apply only for charging with feed water under operational conditions where full feed water temperature is achieved (full power of main turbo-set, charging energy taken from boiler overload). This may not be convenient for operational reasons; in practical cases, there will be an auxiliary preheater to full temperature using live or intermediate steam. Also, during discharge, there will be additional mixing losses if temperature of stored water and feed water temperature are not equal. It is a considerable advantage of the combined steam/feed water flow schemes described above (Fig. 49, 50 and 51) to provide the flexibility necessary to avoid such losses. Storage efficiencies of such combined systems may, therefore, be assumed as intermediate between expansion-type (or, for that matter, displacement) steam storage and the theoretical values of feed water storage and may thus approach or even exceed practical values of feed water storage efficiency.

6. PARAMETRIC STUDY

In order to arrive at optimized plant and power cost figures for a given flow sheet, the following data are required:

- first cost of the reference design peak load plant, consisting of storage related costs (PCIV) and of power dependent costs (remaining items of the peak load plant such as turboset, piping, building, etc.),
- the annual carrying charge,
- cost functions,
- fuel cost of the base load plant,
- thermodynamic data such as specific capacity of accumulator, thermal efficiency, storage efficiency, and finally
- operational data such as maximum discharge capability (load duration) and number of annual discharges.

These data are calculated or assumed in the following paragraphs, and optimizations carried out as applicable.

6.1 Installation cost of reference design PCIV

Based on the reference design of a hot going PCIV storage vessel as described in paragraph 4.3 and the erection sequence as described in paragraph 4.4.9, a cost estimation was carried out. Results are presented in Table XIV. Total cost (\$ 9,984,000 or roughly \$ 10,000,000) includes overheads and similar charges. It does not include:

- transportation (any figure would be arbitrary since neither the location of the foundry nor the site are known);
- internals (an expansion-type storage vessel without internals was assumed; internals of a pressure-drop type storage vessel would increase the total figure by about 3 %);
- contingencies;
- value added tax, import duties etc.

On the other hand, there may be considerable savings due to the following effects:

- Installing more than one PCIV consecutively at the site will lead to cost reductions (for instance, installing three vessels would reduce total cost due to spreading the engineering cost and due to savings in erection cost by about 7 %).
- Choosing the alternative design (horizontal arrangement), keeping L/D ratio, might reduce weight and cost of cast iron as well as of cables, shoes etc. by about 6 %, installation cost by about 5 to 7 %.
- The design may not be fully optimized in terms of unit size and height to diameter ratio L/D. This will apply in particular to the horizontal arrangement.

6.2 Cost functions of PCIV

Based on the reference design, a parameter variation is carried out for the purpose of defining cost functions. The following parameters are varied: operating pressure (1...10 MPa), L/D ratio (1...10), and volume (100...10 000 m³).

6.2.1 Variation of pressure

Weight and cost estimations for various operating pressures are given in Table XV. Results are also plotted in Fig. 54. The curve is only slightly convex, but there is a distinct kink at about 2 MPa. Below this pressure wall thickness is determined by local bending stresses due to the local radial forces of the shoes rather than by tangential compressive stresses.

6.2.2 Variation of L/D ratio

Weight and cost estimations for various height to diameter ratios L/D are presented in Table XVI and Fig. 55. Results show that within the range investigated (L/D up to 10) there is no minimum yet, although there would be a minimum at some higher L/D ratio. It should, however, be noted that higher L/D ratios entail higher thermal insulation losses for fixed insulation thickness due to the higher surface area. Even without this effect, the reduction of installation cost beyond L/D = 6 is rather small.

6.2.3 Variation of PCIV volume

Table XVII presents the results of weight and cost estimations for varying the internal volume of the PCIV. Apart from absolute cost figures, specific figures (\$/m³) are given. These are plotted in Fig. 56 against pressure, with volume as parameter. It turns out that the effect of size is not the same at all pressures. Obviously, for small sizes, full utilization of the wall thickness occurs at higher pressures than for larger units.

6.3 Cost of stored available energy

Whereas the optimization of L/D ratio is a matter of the PCIV only, and also the optimization of PCIV unit volume consists for all practical purposes only in determining the limiting unit volume (above which there is no cost depression or even an increase of specific cost), matters are different with the operating pressure. PCIV cost increases with pressure in a monotonous manner. But increased pressure means higher specific stored electric energy (kW/m³) and thus a smaller vessel. It also means reduced steam mass flow rate and therefore a smaller turbine, pipework, condenser and cooling tower. Optimization of the operating pressure therefore means, for a given flow sheet, minimization of specific PCIV cost (\$/kWh stored) as carried out in this paragraph and, more general, minimization of peak power cost (\$/kWh generated) for a given or chosen value of the annual carrying charge, as carried out in the following two paragraphs.

Optimization of storage pressure was carried out for the following flow sheets:

- (A) Expansion-type steam storage with three flash evaporators, and a separate peak-load turbine:
 - (A 1) with superheating by means of branched-off water (Fig. 47), or
 - (A 2) without superheating,
- (B) Expansion-type feed water and steam storage, with separate peaking turbine arranged in parallel with the main set (Fig. 49) or for overloading the main turboset if possible (Fig. 50), and preferably charged with saturated or slightly superheated steam extracted from the main set at cold reheat and/or cross-over:
 - (B 1) without steam recharging during discharge, or
 - (B 2) with live steam recharging.

According to what has been said in paragraph 4.5 on the use of PCIV for displacement storage vessels, this storage system is not considered for the optimization procedure which is based on the hot going PCIV reference design. Thermodynamically however, types (B 1) and (B 2) correspond below feed water temperature (that is in the examples considered below 255°C or below 41 bar pressure) to a pure feed water storage type. For similar reasons, the varying pressure storage system has been omitted from the following investigations.

Thermal calculation of the expansion-type systems (A) and (B) is somewhat tedious because of the change of conditions during discharge unless recharge is employed. In Table XVIII, the specific hot water output (tonnes/m³) of expansion-type PCIV accumulators for different operating pressures is calculated. Part (i) of Table XVIII concerns the fully charged condition, part ii(a) refers to expansion-type storage without steam recharge during discharge, part ii(b) to recharge at constant pressure with saturated steam, part ii(c) to recharge with throttled live steam.

In Table XIX the electric energy to be gained from 1 tonne of hot water (and its reciprocal, the hot water consumption) with three flash evaporators without superheater is calculated for different operating pressures; it is also shown in Fig. 57 for saturated steam (A 2) and for superheated steam (A 1).

Using the specific hot water output from Table XVIII and the data of Fig. 57, the specific storage volume (m³/kWh) and furthermore the specific PCIV cost (\$/kWh) are calculated in Table XX for expansion-type steam storage without superheating (A 2), in Table XXI for expansion-type feedwater and steam storage without steam recharging (B 1), and in Table XXII for expansion-type feedwater and steam storage with live steam recharging (B 2). Specific PCIV cost-data from Tables XX, XXI and XXII as well as PCIV cost data for expansion-type steam storage with superheating (A 1), based on the data of Fig. 57, are plotted in Fig. 58.

Results of the investigation into the cost of stored available energy may be summarized as follows:

- PCIV cost is higher for case (A) with external steam generation and separate turbine than for case (B) with feed water storage and either overloaded or separate turbine; cost of (A) is higher by 60 % at 20 bar, by 35 % at 60 bar and by 25 % at 100 bar.

- Superheated steam (A 1) yields a small advantage (of a few percent) in PCIV cost against saturated steam (A 2). However, there is additional cost for the external superheater and no, or almost no, net saving. However, superheating may be useful or even required for pressures in excess of 20 bar for reasons of steam wetness, turbine efficiency and for operational reasons (such as start-up).
- Recharging the expansion-type storage vessel during discharge (B 2) has the advantage of constant temperature of the extracted water and therefore constant steam generation and peak power during the discharge period. A disadvantage is the somewhat higher PCIV volume to compensate for the loss of live steam. The use of bleed steam of suitable pressure is theoretically better; it may, however, require an additional bleed nozzle and bleed line.
- The pure expansion-type storage vessel (B 1) is lowest in PCIV cost. Its disadvantage is a slight decrease of pressure, temperature, and additional peaking power during discharge. Such a load profile may sometimes be required anyhow.

6.4 Installation cost of TES peak power plant

As already mentioned, the cost of a TES peak power plant is composed of a power-related portion and an energy-related portion. Cost of the latter may be taken from Fig. 58. Cost of the power-related portion was estimated using several sources [37, 44]; it may be split up again into a part depending on storage pressure or steam pressure and therefore on steam flow rate (turbine, piping, etc.), and another independent part (building, control).

Fig. 59 shows estimated installation cost of a peaking plant with expansion-type steam storage (A) according to Fig. 47 using three evaporator pressure levels, plotted as a function of operating pressure (p_0) for different peak load durations (different size and/or number of storage vessels). Optimum pressure depends strongly on load duration; for very short durations it is about 10 MPa, for 2 hours about 7 MPa, and for 6 hours about 4 MPa.

Fig. 60 shows a similar diagram for combined feed water/steam expansion-type storage (B) with separate peak load turbine according to Fig. 49. Power-dependent cost is similar to Fig. 59, PCIV cost is taken from Fig. 58 (B 1). Total cost is somewhat lower than in Fig. 59. The optimum is at somewhat lower pressures than in Fig. 59, in agreement with the pressure dependence in Fig. 58 (B 1).

In Fig. 61 cost of the combined feed water/steam storage plant (B) with existing overloadable main turboset is shown. Cost for turbine adaption etc. is relatively low, so this is the least expensive plant. However, it may only be applied to existing plants in special cases.

An overall optimization of the operating pressure in combined feed water arrangements (B 1 and B 2) may, however, require more than a pressure optimization according to Fig. 60 or 61. Choosing the operating pressure somewhat higher than the theoretical optimum may be of interest if a high ratio of peak load to base load is required. The curves in Fig. 60 and 61 are rather flat around the optimum, anyway. Fig. 62 shows the additional peak power in percent of the base load power as a function of storage pressure. As an example: Increasing storage pressure from the theoretical optimum 4 MPa for 2 hours discharge to 8 MPa means

an increase of specific cost by 5 % (Fig. 61) but an increase of the possible additional peak power from 14 to 21 % of base load, that is by 50 %. There is not only the advantage of higher peak power but also the cost degression of PCIV and turbine etc. which according to Fig. 56 more than compensates for the 5% cost increase due to higher pressure. Figs. 60 and 61 should therefore be put into the right perspective; they are indicative for pressure dependence but an optimization will have to be carried out for each actual plant or plant type.

A further optimization parameter is condenser pressure of the peaking turbine. Increased condenser pressure will reduce the cost of turbine, condenser, and cooling tower, but will decrease turbine enthalpy drop and therefore increase steam flow rate and cost of storage vessels and piping. Fig. 63 strictly applies to the expansion-type steam storage arrangement without superheat (A 2) and reference design data, but may also be used for different flow sheets and other data as a guide-line. Optimum condenser pressure depends rather strongly on load duration.

Fig. 64 shows the dependence of installation cost of the peak power plant (B 1) on the degree of superheat. The optimum is rather flat. Superheating should therefore be chosen for operational rather than cost considerations as mentioned before.

6.5 Special considerations for solar power stations

The solar power station is different from a fossil fired or nuclear thermal power station in that energy input per design day is given prime consideration rather than power. In a solar power plant with integrated TES it is therefore possible to reduce turbine power from the value corresponding to peak insolation to a value depending on the amount of stored energy (for daily storage). Therefore, turbine power enters the optimization. Three typical cases of the operation of such a plant, whose flow sheet is given in Fig. 52, are schematically shown in Fig. 65 a, b and c /46/.

Fig. 65a (Case I) is the base-load case ("bottoming case"): Constant output may be achieved 24 hours a day. During the night, the low pressure (LP) turbine takes the load; all the steam is taken from the steam storage plant. As the sun rises, the high pressure (HP) part of the turbine is started up with high pressure superheated steam from the solar steam generator; the steam is subsequently used for charging the superheater and steam storage vessel. The LP inlet valve is gradually closed, thereby reducing power output of the LP turbine. In the afternoon the reverse action is taken. The shaded areas determine the necessary storage capacity. The area "LP" is somewhat smaller than the area "Stored Energy" due to the storage efficiency being less than 1.0 (approximately 0.7 to 0.75).

Fig. 65b (Case II) is a somewhat different case, with a power output shape similar to the usual daily demand pattern (larger load during the day, lower load during the night). Contrary to the operation according to Fig. 65a, the LP turbine is now working all the day; its power is somewhat reduced around high noon in accordance with the higher condenser pressure. The HP turbine is operated in a manner similar to the case of Fig. 65a, with the exception that now only part of the steam is used for charging the steam storage vessel, whereas the remainder serves as working steam for the LP part. The LP inlet valve and the storage line inlet valve are now used to control the LP output on the one hand, and the charging operation on the other.

In Fig. 65c (Case III), the turbine power is increased even further than in Case II and the storage volume is decreased accordingly. The mean value of LP turbine power during the night is, therefore, well below the maximum LP power (which, again, is somewhat dependent on ambient temperature). However, power peaks of relatively short duration may be accommodated up to the maximum LP power, as is shown in Fig. 65c by the dotted line in the evening around 8 p.m. (e.g. TV peak).

Fig. 66 shows initial cost versus the "turbine lay-out factor" ψ which is defined as the ratio between maximum turbine power for the mode of operation considered to peak power given by the steam generator without turbine limitation. The following cost data were assumed for the example: 38 \$/kW for the HP turbine, 88 \$/kW for the LP turbine including the cold end (condenser and dry cooling tower). Cost of energy storage was taken as 38 \$/kWh electrical power stored. Cost of the turbine including a dry cooling tower shows a distinct minimum at Case II, which is quite obvious since this is the only case where the expensive LP turbine plus cooling tower is utilized 24 hours a day (Fig. 65).

Cost of the energy storage system by definition is zero at $\psi = 1.0$. It increases almost linearly with $(1 - \psi)$. Assuming a maximum output of the solar steam generator corresponding to 100 MW net electric power and assuming $\psi = 0.44$ (Case II) and therefore a maximum turbine output of $(0.44)(100) = 44$ MW, the required size of the storage vessel, e.g. for Case II, turns out to be 260 MWh, that is approximately one vessel of the size shown in Fig. 29.

Cost of turbine plus storage is also plotted. It is lowest at $\psi = 1.0$ and increases with decreasing ψ , slowly at the beginning and more steeply at low values of ψ . For the above relation of storage cost to turbine cost, initial cost is, therefore, lowest without energy storage (Case IV in Fig. 66), as was to be expected.

6.6 Annual carrying charge

The annual carrying charge is expressed as a percentage of total capital cost of the plant. It includes:

- rate of return (interest),
- depreciation (depending on life time),
- taxes (federal income tax and other taxes),
- construction compound interest factor (CCIF) which accounts for the cost of money during construction.

For present US conditions, the annual carrying charge for utility energy storage installation, with life times of 25 years or more (that is, excluding batteries) is 14 to 15 % /15/.

6.7 Cost of TES peak power

Cost of TES peak power, c , depends on specific capital cost

$$C = C_p + C_s d \quad (6.1)$$

which consists of the power-related part C_p and the storage-related part $C_s d$, and, furthermore, on annual carrying charge a , cost of off-peak power c_{off} , storage efficiency η_{st} , and the utilization which is given by either the annual utilization n or the annual number of discharges $N = n/d$:

$$c = \frac{aC}{n} + \frac{c_{off}}{\eta_{st}} = \frac{a(C_p + C_s d)}{n} + \frac{c_{off}}{\eta_{st}} \quad (6.2)$$

$$c = \frac{aC_p}{n} + \frac{aC_s d}{n} + \frac{c_{off}}{\eta_{st}} = c_1 + c_2 + c_3 = c_{12} + c_3 \quad (6.3)$$

where

$$aC = aC_p + aC_s d \quad (6.4)$$

is the fixed annual cost,

$$c_{12} = \frac{aC}{n} = c_1 + c_2 \quad (6.5)$$

is the installation cost dependent part of peak power cost, and c_3 is the off-peak power related part of peak power cost.

The off-peak power related portion of peak power cost c_3 only depends on fuel cost of the off-peak (base load) plant c_{off} and on storage efficiency η_{st} . Their relation is shown in Fig. 67 where approximate data of base load fuel cost c_{off} for coal-fired, LWR, HWR, HTR, LMFR, fusion and solar power plant are given. As a first approximation, storage efficiency η_{st} may be assumed to be independent of maximum discharge capability d and annual number of discharges N . The two parts of peak power cost, c_3 on the one hand and c_{12} on the other, may therefore be treated independently.

From equation (6.3) it may be seen that the magnitude of c_{12} depends on the annual carrying charge a , the annual utilization n and the specific installation cost of the TES peak power plant $C = C_p + C_s d$. For given values of a and n , the optimum value of c_{12} for a variation of parameters such as storage pressure, condenser pressure, superheating, etc. corresponds to the optimum value of C as per Figs. 59, 60, 61, 63 or 64. For given values of c_{off} and η_{st} , these optimum values of the parameters are also the optimum values regarding the cost of peak power $c = c_{12} + c_3$.

Fig. 68 permits the graphical determination of the cost of peak power ($c_1 + c_2$, without c_3) for a variation of maximum discharge time d , annual number of discharge N , annual utilization n , and annual carrying charge a . The data refer to a plant with combined feed water/steam expansion-type storage and separate peak-load turbine. The diagram consists of three parts.

The left part shows the installation cost of the peak load plant as a function of maximum discharge capability (load duration) d . Data are based on optimum operating pressures p_0 according to Fig. 60 (combined feed water and steam storage).

Annual fixed cost aC of the plant ($\$/\text{kWh}, a$) is calculated in the center part of the diagram for a value of $a = 20 \text{ \%/yr.}$, considered to be the upper limit, vs. annual utilization n , with annual number of discharges $N = n/d$ as parameter. Values of $c_{12} = aC/n$ are determined by the inclination of a beam through the origin.

From the right part of the diagram c_{12} is found for any value of a .

The use of the diagram is illustrated by an example. For $d = 4$ hours, the optimum value of p_0 becomes 4 MPa and with $N = 200$, $n = Nd = 800 \text{ hrs./yr.}$, and $c_{12} = 84 \text{ mills/kWh}$ for $a = 20 \text{ \%/yr.}$ is found. For $a = 15 \text{ \%/yr.}$, $c_{12} = 63 \text{ mills/kWh}$ is finally obtained.

6.8 Time Scale

Further work is required before the concept of steam and/or feedwater storage by the PCIV can be introduced into the power generation market. This work should consist of detailed design studies (for the vertical as well as the horizontal arrangement, and the hot going vessel with cold going tendons as well as the cold going design with internal thermal insulation), a market analysis, further specific optimization work, and the obtaining of approval of the design base from the Authorities concerned.

On the hardware side, a continuation of the experimental work on cast iron fatigue is necessary. For the clarification of some design details the construction and testing of a test module PCIV of, say, 60 m^3 internal volume might be required, to be followed by the construction and testing of a prototype (pilot) plant of about 1000 m^3 internal volume and about 6 MPa working pressure. This size would be just sufficient to demonstrate the feasibility of the 12.0 m diameter flat plates of the 8000 m^3 Reference Design PCIV at full scale; it would, however, result in a very low L/D ratio for the prototype vessel. Or, alternatively, the prototype vessel could be optimized for its size which would result in a diameter of about 6.0 m at a L/D ratio of 6.

Assuming the start of the above work in early 1978, and assuming a minimum time requirement, the time scale could be as follows: The 60 m^3 test module could be built by 1979 and tested by 1980, and the 1000 m^3 prototype vessel could be built by 1982 and tested by 1983. A large unit corresponding to the 8000 m^3 Reference Design could therefore be detail designed in 1983 and built in 1984, and it could commence operation in 1985.

7. PCIV FOR ENERGY STORAGE IN GAS TURBINE PLANTS

7.1 Quick start-up of open cycle gas turbine plants

Apart from energy storage for peak load generation, energy storage is required for instantaneous reserve plants. Considering the large power of modern nuclear units, such thermal or potential energy storage cannot be provided economically for prolonged periods. Therefore, instantaneous reserve plants using energy storage are strictly stop-gap reserve plants. Their duty is to bridge the gap until such time as thermal reserve power plants are started up.

Such thermal reserve power plants used to be written-off steam plants. Today the open cycle gas turbine is in charge for this duty. It has been built in units of 100 MW electric power and more. The gas turbine may be started up rather quickly (minimum start-up time: about one minute /124/). However, its lifetime is severely reduced by such quick start-ups. Therefore, the quick start-up is avoided if at all possible and is practically eliminated. The gas turbine thus loses its stop-gap properties and stop-gap power must be found somewhere else. That is where high-pressure air storage using PCIV may come in.

Fig. 69 shows the arrangement. The basic principle of the method consists in a temperature increase as slow as necessary in order to avoid any marked reduction of life time and, at the same time, an immediate availability of the power output, i.e. a rate of power output increase as fast as the generator can take it. It should be noted that, without the storage system, in the most critical phase power must be taken from the grid for the start-up motor /11/.

The procedure is shown in Fig. 70 for a 100 MW gas turbine plant. Before start-up throttling valve and start-up damper (Fig. 69) are closed; the vent is open. Start-up occurs in the following steps:

- the throttling valve opens; set accelerates; fuel pipe opens; turbine inlet temperature is controlled by flow rate of fuel;
- generator is synchronized;
- throttling valve opens further until rated or maximum power is reached; pressure before turbine is lower than rated because compressor is idling;
- vent closes;
- start-up damper opens slowly until turbine inlet pressure approaches rated value; throttling valve closes slowly, keeping output power constant.

After 10 minutes, full turbine inlet temperature is reached; the start-up damper is fully open, the throttling valve is fully closed; the combustion chamber is on full power; start-up procedure is completed.

Supply pressure at the end of discharge is 8 bar in the example shown. Storage pressure, therefore, is to be much higher. It should be noted that the air storage vessel takes much more air mass at higher pressure. Therefore, high or very high pressures should be used. Also, the vessel should have a large mass, in order to avoid too low a temperature during discharge. Both effects favour the use of the PCIV.

Fig. 71 shows the required specific vessel volume as a function of start-up time and charging pressure. The storage capacity of the PCIV was calculated by using the diagram of Fig. 7. For a 100 MW turbo-set which requires 10 minutes, and 150 bar storage pressure, a volume of 920 m^3 (32 000 cu.ft.) containing $(920)(164)/1000 = 150$ tonnes air is required. Such an air storage PCIV (requiring no thermal insulation) may cost about \$ 3,000,000 or with valves, piping, charging compressor and control about \$ 3,500,000 or 35 \$/kW which is about 20 % of the gas turbine plant cost. Considering the possibilities of quick start-up without lifetime reduction of the gas turbine, this may be a worthwhile expenditure.

7.2 Bottoming in solar closed cycle gas turbine plants

Boeing Engineering and Construction have recently investigated TES for solar power plants based on the Brayton cycle with helium as the working medium /29, 30/. One of the possibilities investigated was sensible heat storage of solids (MgO bricks in a checkerwork arrangement) in an internally insulated welded steel pressure vessel with helium of 816/440°C and 35 bar (550 psi) as the heat transfer agent.

A brief investigation showed that a PCIV would be a more economic solution, in particular if cast iron bricks were used instead of the MgO bricks. Although the cast iron bricks are more expensive than MgO bricks, they require much less - only about 40 % - pressure vessel volume (and, besides, avoid any problem of contamination of the helium cycle).

For the plant with 50 MW electrical net power output, and a storage time (discharge time) of 6 hours extending the full power output from 5 p.m. to 11 p.m., a PCIV volume inside thermal insulation of about 1200 m^3 and a vessel volume outside thermal insulation, inside liner of about 1320 m^3 is required. PCIV dimensions would thus be: 6.5/6.8 m diameter, 36.2/31.5 m height. Weight would be about 1600 tonnes.

7.3 Further applications

Further applications of PCIV for TES in gas turbine plants may include:

- helium storage tanks for load control of the Brayton solar power plant mentioned in paragraph 7.2; and for closed cycle gas turbine plant with helium as working medium generally,
- high-pressure oxygen pressure vessels for thermochemical energy storage (CES) in solar power plants /29, 30/.

8. CONCLUSIONS

From the investigations carried out the following conclusions are drawn regarding feasibility, applications, economics, and development status of the Prestressed Cast Iron Vessel (PCIV) for Thermal Energy Storage (TES) in thermal power plants, in particular steam power plants.

- (1) The envisaged use of the PCIV opens up new technical and economical possibilities of integrated TES for peaking and reserve duty in thermal power plants. Apart from the application in fossil-fired, nuclear, and solar steam plants, use of the PCIV for the quick start-up of peaking gas turbine sets and for sensible heat storage in solar gas turbine plants has been considered.
- (2) High pressure air PCIV for peaking gas turbine start-up would have no thermal insulation at all. PCIV for solar gas turbine high temperature sensible heat storage would have efficient thermal insulation at the inside of the vessel and would therefore be a cold going PCIV. Applications of the PCIV in steam power plants may be either cold going (with inside thermal insulation under water pressure), or hot going with thermal insulation outside the main cast iron body, and the latter one either with cold tendons (necessitating pressurized thermal insulation between shoes and main body) or with hot tendons (of low relaxation or with the possibility of restressing).
- (3) Neither the Codes for steel pressure vessels nor the one developed for concrete vessels can be readily applied to the PCIV. A design base for hot and cold going PCIV is therefore developed and proposed. Its main points are definition of a safety factor $F_g = 1.5$ against gapping and a safety factor $F_t = 1.5$ between tendons failure and gapping. For the hot going vessel with cold going tendons self-stressing is to be considered. Safety against gapping will be higher in the hot condition. Either $F_g = F_{gc} = 1.5$ is applied to the cold condition (resulting in $F_{gh} = 3.86$ for the reference design) or $F_g = F_{gh} = 1.5$ is applied to the hot condition (resulting in $F_{gc} = 0.58$, i.e. pressure testing has to be performed at an elevated temperature, e.g. 125°C); such a procedure is only possible for the varying pressure scheme (with internal steam generation) and the expansion-type scheme but not for displacement storage (where pressure and temperature are not coupled).
- (4) Despite there being no Code available for the PCIV at present, a nuclear pressure vessel (scram drive pressure vessel of 230 bar, 17.5 m³, for the German Thorium High Temperature Reactor) is on order, is being approved by the German TÜV and being built by Siemenskamp KG at Krefeld.
- (5) As is known from Prestressed Concrete Pressure Vessels under development for water-cooled reactors, an efficient thermal insulation inside a vessel with hot water is difficult and expensive (much more so than for a vessel under gas pressure). Particular attention is therefore given to the hot going PCIV. The hot going vessel with hot going tendons needs wires with low relaxation that are not readily available at present. On the other hand, the problems of the pressurized thermal insulation between shoes and main body is solved by using relatively inexpensive "Klingerit 1000" sheets; the hot going vessel/cold going tendons concept was therefore chosen for the reference design that is described, stress analyzed, and costed.

- (6) The reference design PCIV (8000 m³ internal volume, 12 m internal diameter, 71 m internal height, 6 MPa = 60 bar = 900 psi working pressure) provides stored energy corresponding to between about 200 and 300 MW-hours of electric energy and is considerably larger in terms of pressure times volume than any reactor vessel (and probably any pressure vessel) built or considered so far. An alternative reference design, arranged horizontally, should have additional merits regarding weight, cost, and erection procedure.
- (7) Transient thermal stresses during discharge of the reference PCIV have to be considered since they may lead to surface gapping and to fatigue. Surface gapping can be tolerated to some extent and it may be minimized by proper design of the internal surface. Fatigue data are only available for large numbers of cycles (10⁷). Tests with numbers of cycles applicable to TES (10⁴ to 10⁵) are being carried out and should become available during 1977. With present data (for 10⁷ cycles) a temperature shock of only about 40 K and a temperature swing for one hour discharge time of 60 K is possible for the reference design, necessitating the use of the cold going concept for displacement-type storage schemes. However, the expansion-type storage scheme for steam as well as combined feed water and steam storage, developed specifically for the PCIV, only requires temperature swings of 20 K or less and is thus fully compatible with the reference design.
- (8) Optimum pressures depend strongly on the load duration and annual number of discharges considered and vary widely. Typical pressure should be between 40 bar (600 psi) and 80 bar (1200 psi). L/D ratios will be larger than 4 and in many cases even larger than 6. Specific cost should decrease up to unit internal volumes of about 10 000 m³ at high pressures, and about 15 000 m³ at lower pressures.
- (9) Cost of the reference vessel is about \$ 10,000,000 or 33 to 50 \$/kWh of electric energy stored. Capital cost of the peak power plant will be about 50 to 80 \$/kWh electric energy stored. Cost of peak power will be from about 30 to 100 mills/kWh, depending on many parameters.
- (10) Possible further developments of the PCIV for TES applications include:
- different liner concepts and liner materials including the no-liner concept,
 - hot-going tendons with low relaxation or with restressing after a few years of operation,
 - more advanced design stress for cast iron, at least up to 40 % of compressive strength as is permissible according to the ASME Code for concrete reactor vessels, against 25 % of compressive strength used as design base,
 - horizontal arrangement of the PCIV with axial tendons between main body and tangential tendons,
 - use of helically wound tendons covering also the flat plate instead of tangential and axial tendons.
- (11) For the introduction of the TES by PCIV concept into the power generation market, the following procedure is recommended:
- (a) Carry out detailed design studies on the PCIV for hot water as well as for high pressure air and high temperature gas duties, featuring the

vertical as well as the horizontal arrangement, and the hot going vessel with cold going tendons as well as the cold going design with internal thermal insulation (1978/79/80);

- (b) Carry out a market analysis, select suitable reference plants (conventional, nuclear, solar) and optimize the lay-out and design of integrated TES plants for specific cases (1979/80);
 - (c) Continue experimental work on cast iron fatigue data and on some design details; construct and test a 60 m³ test module PCIV (1978/79/80);
 - (d) Obtain approval of the design base from local Authorities (1979/80);
 - (e) Design, order, build and commission a prototype (pilot) plant for steam and for feed water storage of about 1000 m³ internal volume and about 6 MPa = 60 bar (about 900 psi) working pressure (1980/82). Whether the diameter of such a prototype PCIV will be optimized for its size (e.g. 6.0 m for a vessel of 1000 m³ at L/D = 6) or whether it will be the same as for a large vessel (e.g. 12.0 m for a 8000 m³ PCIV), resulting in a very low L/D ratio of the prototype vessel, will have to be decided at a later date.
 - (f) Carry out extended testing of the prototype plant by operating it for one year (1982/83).
- (12) The estimated overall time schedule for the further development of the PCIV for TES is given in item (11) by the years in parenthesis; these are based on a minimum time requirement and on start of work early in 1978 on a large scale.

APPENDIX: LIST OF SYMBOLS USED

A	(m ²)	Cross section of axial tendons
$C = C_p + C_s d$	(\$/kW)	Specific capital cost (installation cost) of storage plant
C_p	(\$/kW)	Power-related part of specific capital cost
C_s	(\$/kWh)	Storage-related part of specific capital cost
D	(m)	Inner diameter (of PCIV)
E	(N/m ²)	Young's modulus
E_c	(N/m ²)	Young's modulus of cast iron
E_{cc}	(N/m ²)	Young's modulus of cast iron, compression
E_{ct}	(N/m ²)	Young's modulus of cast iron, tension
E_s	(N/m ²)	Young's modulus of steel (tendons)
F_f	(-)	Safety factor against tendons failure
F_{freq}	(-)	Required safety factor against tendons failure
F_g	(-)	Safety factor against gapping
F_{gc}	(-)	Safety factor against gapping, cold condition
F_{gh}	(-)	Safety factor against gapping, hot condition
F_{ghreq}	(-)	Required safety factor against gapping at hot condition
F_i	(-)	Factor of inherent safety
G	(kg; tonnes)	Weight
$K=1.2854 \ 1/\sqrt{Rs}$	(-)	Dimensionless function for axial temperature gradient
L	(m)	Internal height (length) of PCIV
$N = n/d$	(1/a)	Annual number of discharges
R	(m)	Inner radius (of PCIV)
S	(N/m ²)	Stress intensity according to ASME Code, Section III
S_m	(N/m ²)	Design stress intensity according to ASME Code, Section III
T_u	(°C)	Upper temperature
T_l	(°C)	Lower temperature
T_i	(°C)	Initial temperature
ΔT	(K)	Temperature difference; temperature ramp height
ΔT_{eq}	(K)	Equivalent temperature difference
V	(m ³)	Inner volume of PCIV
W	(kJ/K)	Heat capacity

W_o	(kJ/K)	Heat capacity of water content
W_{Fe}	(kJ/K)	Heat capacity of iron wall
a	(%/yr.)	Annual carrying charge
$a = \lambda/\rho c$	(m ² /s)	Temperature conductance
a_{eff}	(m ² /s)	Effective temperature conductance
$b = dT/dt$	(K/s)	Temperature rise
c	(kJ/kgK)	Specific heat capacity
$c = c_{12} + c_3$	(\$/kWh)	Peak power cost
$c_1 = a C_p/n$	(\$/kWh)	Part of peak power cost related to C_p
$c_2 = a C_s d/n$	(\$/kWh)	Part of peak power cost related to C_s
$c_{12} = c_1 + c_2$	(\$/kWh)	Installation cost dependent part of peak power cost
$c_3 = c_{off}/n_{st}$	(\$/kWh)	Off-peak power related cost of peak power cost
c_{off}	(\$/kWh)	Off-peak power cost (fuel cost)
$d = n/N$	(hrs.)	Maximum discharge time (discharge capability)
f_c	(N/m ²)	Design stress for cast iron
f_{cc}	(N/m ²)	Design compressive stress for cast iron
f_{ct}	(N/m ²)	Design tensile stress for cast iron
f_s	(N/m ²)	Design stress for steel tendons
f_{sf}	(N/m ²)	Design stress for steel tendons at failure conditions
h	(kJ/kg)	Enthalpy
h_p	(m)	Height of the flat plate
h_{po}	(m)	Outer height of the flat plate
h_{pi}	(m)	Inner height of the flat plate
l	(m)	Mixing length
$n = Nd$	(hrs./yr.)	Annual utilization
p	(N/m ²)	Pressure
p_i	(N/m ²)	Internal design pressure
p_o	(N/m ²)	Internal operating pressure
p_1	(N/m ²)	Upper operating pressure
p_2	(N/m ²)	Lower operating pressure
s_c	(m)	Wall thickness of cast iron, cylindrical part
s_{cp}	(m)	Thickness of cast iron, flat plate
s_{cpeq}	(m)	Equivalent thickness of girder type flat plate

s_s	(m)	Thickness of steel tendons, cylindrical part, considered as a solid ring
s_{sp}	(m)	Thickness of steel tendons, flat plate, considered as a solid ring
t	(hrs.; s)	Time
$t^+ = \frac{\lambda}{\rho c} \frac{1}{s^2} t$	(-)	Dimensionless time (Fourier number)
Δt	(hrs.; s)	Transition time
t_d	(hrs.; s)	Charging (discharging) time
α	(W/m ² K)	Heat transfer coefficient
α_r	(W/m ² K)	Radiative heat transfer coefficient
α_c	(W/m ² K)	Convective heat transfer coefficient
$\alpha^* = \alpha s / \lambda$	(-)	Dimensionless heat transfer (Biot Number)
β	(-)	Coefficient of thermal expansion
$\epsilon_1, \epsilon_2, \epsilon_3$	(%)	Strain components in direction of the principal stress components $\sigma_1, \sigma_2, \sigma_3$
ϵ_r	(%)	Radial strain
ϵ_t	(%)	Tangential (circumferential) strain
ϵ_x	(%)	Axial strain
ϵ_g	(%)	Strain of tangential cables at gapping condition
ϵ_{gc}	(%)	Strain of tangential cables at gapping condition, cold state
ϵ_{gh}	(%)	Strain of tangential cables at gapping condition, hot state
ϵ_i	(%)	Strain of tangential cables at design pressure
ϵ_{max}	(%)	Fictitious elastic strain of tangential cables at failure conditions
ϵ_p	(%)	Strain of tangential cables at prestressing condition
ϵ_{pc}	(%)	Strain of tangential cables at prestressing condition, cold state
ϵ_{ph}	(%)	Strain of tangential cables at prestressing condition, hot state
η_{st}	(-; %)	Storage efficiency
λ	(W/mK)	Thermal conductivity
λ_{eff}	(W/mK)	Effective thermal conductivity (of hollow plate)
ν	(-)	Poisson's ratio
ρ	(kg/m ³)	Density

σ	(N/m ²)	Stress or strength
$\sigma_1, \sigma_2, \sigma_3$	(N/m ²)	Principal stress components
σ_b	(N/m ²)	Bending stress or strength
σ_c	(N/m ²)	Stress or strength for cast iron
σ_{cc}	(N/m ²)	Compressive stress, cast iron
σ_{ct}	(N/m ²)	Tensile stress, cast iron
σ_{cuc}	(N/m ²)	Ultimate compressive strength, cast iron
σ_{cut}	(N/m ²)	Ultimate tensile strength, cast iron
σ_i	(N/m ²)	Stress at the inside of the flat plate
σ_{per}	(N/m ²)	Permissible stress
$\sigma_r, \sigma_t, \sigma_x$	(N/m ²)	Radial, tangential (circumferencial) and axial stress component
σ_o	(N/m ²)	Stress at the outside of the flat plate
σ_s	(N/m ²)	Stress or strength for steel
σ_{st}	(N/m ²)	Tensile stress, steel
σ_{sut}	(N/m ²)	Ultimate tensile strength, steel
σ_{sy}	(N/m ²)	Yield stress, steel
$\sigma_{s0.2}$	(N/m ²)	Yield stress (0.2 % plastic strain), steel
σ^+	(-)	Dimensionless thermal stress
σ_F^+	(-)	Dimensionless thermal stress, temperature shock
σ_r^+	(-)	Dimensionless thermal stress, temperature ramp
τ	(-)	Shear stress
ψ	(-)	Turbine lay-out factor

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TABLE 1: THERMAL ENERGY STORAGE SYSTEMS

Type of thermal energy	sensible					latent			
						phase change	heat of formation		
Medium	solid	liquid				molten salt (e.g. eutectic metal fluorides)	solid/gas	liquid/gas	gas/gas
		high boiling			H ₂ O		metal hydrides, salt-NH ₃ etc.	H ₂ O-NH ₃	CO+3 H ₂ ⇌ CH ₄ +H ₂ O
		liquid metal	HITEC	oil or other organic liquid					
Temperature °C	0...1500	100...1000	150...600	50...400	0...350	450...850	0...500 (900)	0...300	600...900
Pressure MPa	0...10	0...1	0...1	0...3	0.1...16	0...1	1...5	0.1...10	1...2
Corrosion resistant vessel required	no	yes	yes	no	no	yes	no	yes	no

TABLE II: LOSSES AND STORAGE EFFICIENCIES OF ENERGY STORAGE SYSTEMS

Storage system			Losses of available energy			Relative energy requirement	Storage efficiency
			charging	accumulator	discharging		
Hydraulic pumped storage			electric mains, pump efficiencies	seepage, evaporation	pipe friction, turboset efficiencies, exit	1.33...2.0(e)	0.5 ...0.75
Air storage with gas turbines	varying pressure type		electric mains, compressor, intermediate and after cooling, pipe friction	loss of heat and pressure, leakage	pipe friction, throttling, turboset efficiencies, flue gas	0.8...0.9(e) 1.25...1.66(th) 2.05...2.56	(1.11...1.25) (0.6...0.8) 0.39...0.49
	displacement type		-	leakage	pipe friction, turboset efficiencies, flue gas	0.75...0.85(e) 1.18...1.5(th) 1.93...2.35	(1.18...1.33) (0.67...0.85) 0.43...0.52
Pressurized hot water storage	internal steam generation (varying pressure type)		pipe friction, throttling	thermal insulation	throttling, efficiency difference of turboset, condenser pressure increase	1.33...2.0(e)	0.5 ...0.75
	external steam generation	displacement type	heat transfer, recirculation power	thermal insulation, mixing ⁺)	heat transfer or throttling, efficiency difference of turboset, recirculation power	1.18...1.43(e)	0.7 ...0.85
		expansion type	heat transfer or mixing, pumping power	thermal insulation	heat transfer or throttling, efficiency difference of turboset, condenser pressure increase	1.18...1.43(e)	0.7 ...0.85
	feedwater storage	displacement type	recirculation power	thermal insulation, mixing ⁺)	efficiency difference of turboset, recirculation power, condenser pressure increase	1.05...1.33(e)	0.75...0.95
		expansion type	pumping power	thermal insulation	efficiency difference of turboset, condenser pressure increase	1.0 ...1.25(e)	0.8 ...0.95
		indirect, with high boiling oil	heat transfer, pumping power	thermal insulation	heat transfer, efficiency difference of turboset, condenser pressure increase, pumping power	1.33...1.67(e)	0.6 ...0.75

⁺) only in case of partially charged state

TABLE III: MATERIAL PROPERTIES OF GREY CAST IRON

Material specification /84, 125/: SKM-U84 (Siempelkamp trade mark)¹⁾

Property	Symbol	Units	Value	
			20 ^o C	300 ^o C
(a) <u>Strength values</u>				
Ultimate tensile strength	σ_{cut}	N/mm ²	200 ²⁾	180
Ultimate compression strength	σ_{cuc}	"	800	640
Compressive yield strength	-	"	400	320
Bending strength	σ_b	"	400	320
Shear strength	τ	"	250	210
(b) <u>Design Stresses (stress limits)</u>				
tensile, primary	f_{ct}	N/mm ²	0	0
compressive, primary	f_{cc}	"	200	160
tensile, local peak	-	"	50	40
compressive, local peak	-	"	267	213
(c) <u>Ultimate strain</u>				
tensile	δ_t	%	0.5	0.8
compression	δ_c	%	20	20
(d) <u>Young's modulus</u>				
compression	E_{cc}	N/mm ²	100 000	90 000
tension at 40 MPa	E_{ct}	"	100 000	90 000
" " 100 MPa	E_{ct}	"	95 000	85 000
(e) <u>Physical properties</u>				
Poisson's ratio	ν	-	0.28	0.28
Thermal expansion	β	K ⁻¹	(9.5)10 ⁻⁶	(12.2)10 ⁻⁶
Thermal conductivity	λ	W/mK	50	47
Specific heat capacity	c	kJ/kgK	0.48	0.59
Density	ρ	kg/m ³	7200	7140

1) SKM-U84 is in accordance with GG-20 (Material No. 0.6020) of German standard DIN 1691 /102/.

2) According to DIN 1691 /102/ dependent on wall thickness; for SKM-U84 valid up to 300 mm wall thickness

TABLE IV: MATERIAL PROPERTIES OF TENDONS

Material specification: St 160 ... 180, cold drawn;

DYFORM L-R, etc. /113, 115, 116/

Composition: C = 0.83 %, Si = 0.27 %, Mn = 0.63 %, P ≤ 0.026 %, S ≤ 0.034 %

Property	Symbol	Unit	Value			
			20°C	60°C	100°C	300°C
(a) <u>Strength values</u>						
Ultimate tensile strength	σ_{sut}	N/mm ²	1720	1700	1680	1190
Yield strength 0.2 %	$\sigma_{s0.2}$	"	1540	1490	1400	850
" " 0.03 %	$\sigma_{s0.03}$	"	1420	1260	1210	580
" " 0.01 %	$\sigma_{s0.01}$	"	1340	-	-	-
(b) <u>Design Stresses</u> (stress limits)						
construction	f_s	N/mm ²	1200	-	-	520
failure	f_{sf}	"	1720	1700	1680	1190
(c) <u>Ultimate strain</u>						
Average	δ		2.0	?	?	?
Related to L = 50	δ_5		6.0	?	?	?
(d) <u>Creep rate at 0.7 σ_{sut}</u>						
after 1000 hrs.	-	%	0.006	0.02	0.045	-
" 10 000 hrs.	-	"	0.009	0.029	0.06	-
" 100 000 hrs.	-	"	0.013	0.04	0.08	-
" 1 000 000 hrs.	-	"	0.017	0.05	0.1	-
(e) <u>Relaxation</u> (1000 hrs.)						
at 0.85 σ_{sut}	-	% ⁺⁾	5.5	12	16	?
" 0.8 σ_{sut}	-	"	2.5	8	12	?
" 0.7 σ_{sut}	-	"	1.2	3.1	6.6	?
" 0.6 σ_{sut}	-	"	0.9	2.6	4.7	?
" 0.5 σ_{sut}	-	"	0.7	2.4	3.7	?
(f) <u>Young's modulus</u>						
	E_s	N/mm ²	205000		180500	
(g) <u>Physical properties</u>						
Poisson's ratio	ν	-	0.3		0.3	
Coefficient of thermal expansion	β	K ⁻¹	(11.1)10 ⁻⁶		?	
Density	ρ	kg/m ³	7850		7800	

+) of initial load

TABLE V: WORKED EXAMPLE OF COLD GOING PCIV

<u>(a) Size and operating conditions</u>			
Internal radius	$R = 6$	m	
" height	$L = 71$	m	
operating pressure (steam cushion)	$p_o = 6$	MPa	
" temperature	275	$^{\circ}\text{C}$	
<u>(b) General design data</u>			
Design temperature, tendons	60	$^{\circ}\text{C}$	
" " , cast iron	60	$^{\circ}\text{C}^{+}$	
" pressure $p_i = 6 + 0.7 + 0.1 =$	6.8	MPa	(4.5a)
Safety factors:	$F_f = 2.25$		
	$F_i = 1.5$		
	$F_g = 1.5$		
<u>(c) Design data of cylindrical part</u>			
$\epsilon_{\max} = f_{sf}/E_s = 1700/210\,000 = 0.008095$			(4.2)
$\epsilon_g = \epsilon_{\max}/F_i = 0.008095/1.5 = 0.005397$			(4.3)
$\epsilon_p = \epsilon_g - f_{cc} (1 - \nu/2)/E_c = 0.005397 - (200)(0.86)/100\,000 = 0.003677$			(4.4)
$s_s = F_f R p_i / f_{sf} = (2.25)(6)(6.8)/1700 = 0.054$	m		(4.5)
$s_c = \epsilon_p s_s E_s / f_{cc} = (0.003677)(0.054)(210\,000)/200 = 0.208$	m		(4.6)
$A = S_f R^2 \pi p_i / f_{sf} = (2.25)(6^2)\pi(6.8)/1700 = 1.018$	m^2		(4.7)
<u>(d) Design data of flat plates</u>			
$p_g = F_g p_i = (1.5)(6.8) = 10.2$	N/mm ²		(4.8)
$\sigma_b = f_{cc}/2 = 200/2 = 100$	N/mm ²		(4.9)
$s_{gb} = \sqrt{1.23 p_g R^2 / \sigma_b} = \sqrt{(1.23)(10.2)(6^2)/100} = 2.13$	m		(4.13a)
$h_{po} = 3.2$ m $h_{pi} = 2.6$ m			(4.13b)
$s_{cpeq} = \sqrt{(h_{po}^3 - h_{pi}^3)/h_{po}} = \sqrt{(3.2^3 - 2.6^3)/3.2} = 2.18$	m		(4.14)
$s_{sb} = F_i (R + s_c) \sigma_b / f_{sf} = (1.5)(6.208)100/1700 = 0.548$	m		(4.14)

⁺) Values practically identical to values at 20 $^{\circ}\text{C}$

TABLE VI: WORKED EXAMPLE OF HOT GOING PCIV, CONSERVATIVE DESIGN BASE

<u>(a) Size and operating conditions</u>			
Internal radius	$R = 6$	m	
" height	$L = 71$	m	
operating pressure (steam cushion)	$p_o = 6$	MPa	
" temperature	275	$^{\circ}\text{C}$	
<u>(b) General design data</u>			
Design temperature, tendons	60	$^{\circ}\text{C}$	
" " cast iron	277	$^{\circ}\text{C}$	
" pressure	$p_i = 6 + 0.7 + 0.1 = 6.8$	MPa	(4.5a)
Safety factors	$F_i = 1.5$		
	$F_{gc} = 1.5$		
<u>(c) Design data of cylindrical part</u>			
$\epsilon_{\max} = 1700/210\,000 = 0.008095$			(4.2)
$\epsilon_{gh} = 0.008095/1.5 = 0.005397$			(4.15)
$\epsilon_{gc} = 0.005397 - (12) 10^{-6} (275) = 0.002097$			(4.16)
$\epsilon_{ph} = 0.005397 - (164/86\,250)(0.86) = 0.003762$			(4.17)
$s_s = \frac{(1.5)(6)(6.8)}{(1700)(0.002097)/0.008095} = 0.139\text{ m}$			(4.20)
$F_f = \frac{(0.139)(1700)}{(6)(6.8)} = 5.79 > 3$			(4.21)
$F_{gh} = 5.79/1.5 = 3.86 > 2$			(4.22)
$s_s = (0.003762)(0.139)(210\,000)/164 = 0.670\text{ m}$			(4.23)
$\delta = (5.79)(6^2)\pi(6.8)/1700 = 2.619\text{ m}^2$			(4.7)
<u>(d) Design data of flat plates</u>			
$p_g = (3.86)(6.8) = 26.25\text{ N/mm}^2$			(4.25)
$\sigma_b = 164/2 = 82\text{ N/mm}^2$			(4.9)
$s_{cp} = \sqrt{1.23(26.25)(6^2)/82} = 3.76\text{ m}$			(4.13a)
$h_{cp} = 4.5\text{ m}$ $h_{ol} = 3.0\text{ m}$			
$s_{cpeq} = \sqrt{(4.5^3 - 3^3)/4.5} = 3.77 > 3.76$			(4.13b)
$s_{sp} = 1.5(6.67)82/1700 = 0.483\text{ m}$			(4.14)

TABLE VII: WORKED EXAMPLE OF HOT GOING PCIV, ADVANCED DESIGN BASE

(a) Size and operating conditions

Internal radius	$R = 6$	m
" height	$L = 71$	m
operating pressure (steam cushion) p_o	$= 6$	MPa
" temperature	275	$^{\circ}\text{C}$

(b) General design data

Design temperature, tendons	60	$^{\circ}\text{C}$
" " cast iron	277	$^{\circ}\text{C}$
" pressure p_i	$= 6 + 0.7 + 0.1 = 6.8$	MPa
Safety factors: F_i	$= 1.5$	
F_{gh}	$= 1.5$	
F_{gc}	$= 1.5 (1 - \frac{(12)10^{-6}(275)}{0.008095/1.5}) = 0.58$	(4.24)

(c) Design data of cylindrical part

$\epsilon_{\max}$	$= 0.008095$	(4.2)
ϵ_{gh}	$= 0.005397$	(4.15)
ϵ_{gc}	$= 0.002097$	(4.16)
ϵ_{ph}	$= 0.003762$	(4.17)
ϵ_i	$= 0.003762 + (0.005397 - 0.003762)/1.5 = 0.004852$	(4.18)
$\sigma(\epsilon_i)$	$= (210\ 000)(0.004852) = 1019\ \text{N/mm}^2$	(4.19)
$s_{\bar{s}}$	$= \frac{(0.58)(6)(6.8)}{(1700)(0.002097)/0.008095} = 0.054\ \text{m}$	(4.20)
F_f	$= \frac{(0.054)(1700)}{(6)(6.8)} = 2.25$	(4.21)
$s_{\bar{c}}$	$= (0.003762)(0.054)(210\ 000)/164 = 0.260\ \text{m}$	(4.23)
A	$= (2.25)(6^2)\pi(6.8)/1700 = 1.018\ \text{m}^2$	(4.7)

(d) Design data of flat plate

p_g	$= (1.5)(6.8) = 10.2\ \text{N/mm}^2$	(4.25)
σ_b	$= 164/2 = 82\ \text{N/mm}^2$	(4.9)
$s_{\bar{b}}$	$= \sqrt{(1.23)(10.2)(6^2)/82} = 2.347\ \text{m}$	(4.13a)
$h_{\bar{b}}$	$= 3.56\ \text{m}$	
$h_{\bar{p}_i}$	$= 2.92\ \text{m}$	
s_{cpeq}	$= \sqrt{(3.56^3 - 2.92^3)/3.56} = 2.38\ \text{m} > 2.347$	(4.13b)
$s_{\bar{d}}$	$= (1.5)(6.26)(82)/1700 = 0.453\ \text{m}$	(4.14)

TABLE VIII: OPERATIONAL DATA OF REFERENCE DESIGN (hot going PCIV with cold going cables)

(a) Pressure	
Operating pressure (steam cushion), p_o	60 bar
Static height	71 m
Density of water	760 kg/m ³
Static head	5.3 bar
Operating pressure, bottom	65.3 bar
Design pressure, p_i	68 bar
(b) Temperature	
Operating temperature	275 °C
Design " "	277 °C
(c) Capacity (approx., for 8000 m³ vessel)	
(c1) used as varying pressure accumulator	
Steam cushion, fractional volume	15 %
Water content, " "	85 %
Water content	6800 m ³ = 5170 tonnes
Specific capacity (electric energy):	27 kWh/m ³ = 97.2 MJ/m ³ (Fig.8, C1)
total capacity (electric energy):	(27)(8000)/1000 = 216 MWh = 0.78 TJ
(c2) used as expansion-type accumulators in a feed water and steam storage system :	
Steam cushion, fractional volume	5 %
Water content " "	95 %
Water content	7600 m ³ = 5780 tonnes
Specific capacity (electric energy):	41.6 kWh/m ³ = 150 MJ/m ³ (Table XXII)
total capacity:	(41.6)(8000)/1000 = 333 MWh = 1.2 TJ
(d) Storage efficiencies	
(d1) used as varying pressure type accumulator	
Storage efficiency (without heat loss), approx.	0.70
" " , with 24 hrs. heat loss, approx.	0.69
(d2) used as expansion-type accumulator in a feed water and steam storage system	
Storage efficiency (without heat loss), approx.	0.86
" " with 24 hrs. heat loss, approx.	0.85

TABLE IX : MAIN DESIGN DATA OF REFERENCE PCIV (hot going PCIV with cold going cables)

(1) Overall data	
internal diameter, D	12.0 m
internal height, L	71.0 m
internal volume, V	8000 m ³
cylindrical wall thickness, s _c ¹⁾	0.26 m
thickness of flat plate (max.) h _{po} ¹⁾	3.56 m
external diameter (cast iron), 12 + 2(0.26) =	12.52 m
external height (cast iron), 71 + 2(3.56) =	78.12 m
external diameter (cables), max.	14.00 m
(2) Cast iron blocks	
(2.1) Cylindrical part	
height of rings	2.0 m
number of rings	35
number of blocks per ring	6
number of blocks	210
(2.2) Flat plates	
number of blocks per plate, 6 + 6 + 6 + 6 + 6 + 3 + 1 =	34
thickness of outer and inner walls	0.32 m
" " intermediate walls, horizontal	0.08 m
" " " " , vertical	0.06 m
At point of maximum thickness:	
outer girder plate	0.32 m
hollow space	0.92 m
intermediate wall	0.08 m
hollow space	0.92 m
intermediate wall	0.08 m
hollow space	0.92 m
inner girder plate	0.32 m
overall height	3.56 m

(continued)

1) see Table VII

TABLE IX (continued)

(3) Prestressing system(3.1) General

Cable diameter	7.0	mm
" cross section	38.5	mm ²
" weight per m	0.302	kg/m

(3.2) Axial cables

required cross-section, ideal case, A^1	1.018	m ²
efficiency factor	0.85	
cross section, effective: $1.018/0.85 =$	1.20	m ²
required number of cables: $(1.2)10^6/38.5 =$	31 104	
number of tendons	72	
" of cables per tendon	432	
diameter of the tendon bundle	155	mm
cable length, total: $(31\ 104)(80)(10^{-3}) =$ approx.	2500	km

(3.3) Tangential cables, cylindrical part

cross-section per m height, ideal case, s_s^1	0.054	m ² /m
efficiency factor	0.85	
cross-section per m height, eff.: $0.054/0.85 =$	0.064	m ² /m
" " total: $(0.064)(71) =$	4.5	m ²
number of cables per m height: $(0.064)10^6/38.5 =$	1663	
height of cable bundles	0.7	m
clearance of cable bundles	0.3	m
pitch " " "	1.0	m
number of cables per layer	100	
" of layers, $1663/100 =$	16.7	
radial cable pitch	8.5	mm
cable bundle thickness: $(16.7)(8.5)10^{-3} =$	140	mm
internal diameter of cable bundle	13.0	m
external " " " "	13.28	m
cable length, total: $(13.14)(1663)(71)(10^{-3}) =$	4900	km

(3.4) Tangential cables, upper and lower flat plate

cross-section per m height, ideal case, s_{sp}^1	0.453	m ² /m
efficiency factor	0.85	
cross-section per m height, eff.: $0.453/0.85$	0.533	m ² /m

1) see Table VII.

TABLE IX (continued)

total height of outer and inner walls: $(4)(0.32) =$	1.28	m
cross-section, total: $(0.533)(1.28) =$	0.682	m ²
number of cables, total	17 720	
" " " per flange: $17\,720/4 =$	4 430	
height of cable bundle	0.7	m
number of cables per layer	100	
" " layers: $4430/100 =$	44.3	
radial cable pitch	8.5	mm
cable bundle thickness: $(44.3)(8.5)10^{-3}$	0.377	m
internal diameter of cable bundle	13.0	m
external " " " "	13.754	m
cable length, total: $(13.377) \pi (17\,720)(10^{-3}) =$	750	km
equivalent height		
flat plates: $(4)(44.3/16.7)(13.377/13.14) =$	10.8	m
cylinder	71.0	m
total	81.8	m
(3.5) Shoes		
Number of shoes in tangential direction, cylindrical part	36	
angle of directional change, cylindrical part: $360/36 =$	10^0	
axial pitch, cylindrical part	1.0	
number of shoes in tangential direction, flat plates	72	
angle of directional change, flat plates: $360/72 =$	5^0	
number of shoes, cylindrical part: $71 \times 36 =$	2556	
" " " flat plates: $4 \times 72 =$	288	
" " " total	2844	
cable tension, cylindrical part ²⁾ : $(1663)(38.5)(1019) =$	65.2	MN
radial force per shoe: $(2)(65.2) \sin (10^0/2) =$	11.37	MN
cable tension, flat plate: $(4430)(38.5)(1019) =$	174	MN
radial force per shoe: $(2)(174) \sin (5^0/2) =$	15.18	MN
tension of axial tendon ²⁾ : $(432)(38.5)(1019) =$	16.95	MN

1) see Table VII

2) tensile stress in cables at operational conditions, see Table VII

TABLE IX (continued)

(3.6) Thermal insulation (Pressurized)

tangential:

area per shoes, cylindrical part:	$(900)(130) =$	117 000	mm ²
compressive stress:	$(11.37)(10^6)/117\ 000$	97.2	MPa
total area under shoes, cylindrical part:	$(117\ 000)(10^{-6})(2556) =$	299	m ²
area per shoe, flat plate:	$(900)(175) =$	157 500	mm ²
compressive stress:	$(15.18)(10^6)/157\ 500 =$	96.4	MPa
total area under shoes, flat plate:	$(157\ 500)(10^{-6})(288) =$	45.5	m ²

axial:

area per tendon head:	$(500^2 - 160^2)\pi/4 =$	176 243	mm ²
compressive stress:	$(16.95)(10^6)/176\ 243 =$	96.2	MPa
total area under tendon heads:	$(176\ 243)(10^{-6})(72)(2) =$	25.5	m ²
number of Klingerit sheets		54	
thickness of Klingerit, uncompressed:	$(54)(3.0) =$	162	mm
" " " compressed:	$(54)(2.76) =$	150	mm
total thickness		176	mm

(4) Liner

Area cylindrical part		2677	m ²
" flat plates	$2 \times 113 =$	226	m ²
" total		2903	m ²
Wall thickness		1	mm

TABLE X: WEIGHT ESTIMATE OF REFERENCE PCIV (hot going PCIV with cold going cables)

(a) <u>Cast iron</u> (including keys, penetrations, support)			
cylindrical part:	$(210)(24.3) =$	5100	
flat plates:	$(2)(34)(30.3) =$	<u>2060</u>	
	total		7160 tonnes
(b) <u>Cables</u>			
tangential, cylindrical part	$(4900)(0.302) =$	1480	
" flat plates	$(750)(0.302) =$	226	
axial:	$(2500)(0.302) =$	<u>755</u>	
	total $(8150)(0.302) =$		2461 tonnes
(c) <u>Cables heads</u>			24 tonnes
(d) <u>Shoes</u>			
cylindrical part:	$(2556)(140) =$	350	
flat plates:	$(288)(320) =$	92	
miscellaneous:		<u>8</u>	
	total =		450 tonnes
(e) <u>Thermal insulation</u>			
under shoes and tendon heads:	$(370)(0.152)(2000)(10^{-3}) =$	112	
others:		<u>8</u>	
	total		120 tonnes
(f) <u>Liner</u>			25 tonnes
(g) <u>Internals</u> (only for internal steam generation: 60 tonnes)			-
(h) <u>Miscellaneous</u> (outer cover, craneage, etc.):			480 tonnes
	total (a) to (h)		<u>10720 tonnes</u>

TABLE XI : THERMAL INSULATION PROPERTIES OF PRESSURE-RESISTANT MATERIALS

	Compressive strength σ N/mm ²	Permissible compressive stress σ_{per} N/mm ²	Thermal conductivity $\lambda(150^\circ\text{C})$ W/mK	Ratio $\lambda/\sigma_{\text{per}}$ W/mKPa
Alumina: RAPAL 85	1700	(1130)	8.2	(0.0073)
" 90	2000	(1330)	9.2	(0.0069)
" 95	2400	(1600)	10.5	(0.0066)
" 99	3000	(2000)	18.8	(0.0094)
" 100	3200	(2130)	25.1	(0.0118)
Isographite (SIGRAflex): $\rho = 1000 \text{ kg/m}^3$	4	4	7	1.75
$\rho = 1500 \text{ " "}$	20	20	5	0.25
(+) $\rho = 2000 \text{ " "}$	200	200	4	0.025
(+) $\rho = 2300 \text{ " "}$	667	(667)	3.3	(0.005)
(+) $\rho = 2500 \text{ " "}$	1000	(1000)	3	(0.003)
Klingerit 1000	100	100	0.4	<u>0.004</u>
Cast iron	656	219 ⁺⁺)	50	0.23
Stainless steel	1500	(1000) 219 ⁺⁺⁺)	20 20	(0.020) 0.09

^{*)} values to be confirmed

⁺⁺⁾ permissible compressive stress: $656/3 = 219$

⁺⁺⁺⁾ This low value is determined by the permissible compressive stress of cast iron

TABLE XII: MATERIAL PROPERTIES OF PRESSURIZED THERMAL INSULATION

Material specification: Rubber asbestos with metal wire mesh /126/

Trade mark: Klingerit 1000

Supplier: Canadian Klinger Ltd.

1596 Oulette Ave.

Windsor 14, Ontario, Canada

Property	Symbol	Unit	Value	
			20°C	300°C
Thickness	s_0	mm	3.0	3.0
Density (uncompressed)	ρ	kg/m ³	2000	-
Permissible compression	σ	N/mm ²	100	-
Compressibility in accordance with ASTM F 36	-	-	0.08	-
Thickness after compression	s	mm	2.76	-
Thermal conductivity	λ	W/mK	0.4	0.4
Maximum operating temperature	T	°C	500	

TABLE XIII: HEAT LOSSES

(1) Shoes and tendon-heads		
Effective area (Table IX, 3.6): $A = 299 + 45.5 + 25.5 =$	370	m ²
Temperature difference: $\Delta t = (275 - 25) =$	250	K
Thermal conductivity λ of Klingerit (Table XII)	0.4	W/mK
Effective thickness s (Table IX, 3.6)	0.15	m
$\lambda/s = 0.4/0.15 =$	2.7	W/m ² K
Heat transfer coefficient, outer (cables), approx.	100	"
" " " inner, approx.	1000	"
" " " overall	2.6	"
Heat flux: $q = (2.6)(250)$	650	W/m ²
Heat loss: $Aq = (370)(650)/1000 =$	<u>241</u>	<u>kW</u>
(2) Remaining surface		
Total area: $3073 + (2)(123) =$	3319	m ²
Remaining area: $A = 3319 - 370 =$	2949	m ²
Temperature difference	250	K
Thermal conductivity λ of fiber insulation	0.04	W/m ² K
Effective thickness s " " "	0.2	m
$\lambda/s = 0.04/0.2$	0.2	W/m ² K
Overall heat transfer coefficient	0.196	"
Heat flux, q	49	W/m ²
Heat loss: $Aq = (2949)(49)/1000 =$	<u>145</u>	<u>kW</u>
(3) Total		
Shoes and tendon-ends	241	kW
Remaining surface	145	"
Piping penetrations etc., say	14	"
	<u>400</u>	<u>kW</u>

TABLE XIV: COST ESTIMATION FOR REFERENCE DESIGN

(hot going PCIV with cold going cables)

Main data (reference size): $V = 8000 \text{ m}^3$, $p_0 = 6 \text{ MPa}$, $D = 12 \text{ m}$, $L = 71 \text{ m}$

Cost base: January 1977

Cost item	Quantity or number of units	Unit cost \$	Itemized cost \$
(a) Cast iron ⁺ , cylindrical part flat plates	5 100 000 2 060 000 <hr/> 7 160 000 kg	0.6	4 296 000
(b) Cables, tangential, cylindrical part flat plates axial	1 480 000 226 000 755 000 <hr/> 2 461 000 kg	0.88	2 166 000
(c) Cable heads	31 104	4.0	124 000
(d) Shoes	450 000 kg	0.8	360 000
(e) Shoe insulation	120 000 kg	3.0	360 000
(f) Remaining insulation	2 557 m ³	20.0	51 000
(g) Liner	25 000 kg	6.4	160 000
(h) Internals	-	-	-
(i) Outer cover, crange, miscellaneous	5 500 m ²	50.0	275 000
SUBTOTAL (a) to (i)			<u>7 792 000</u>
(j) Engineering ⁺⁺			300 000
(k) Erection (21 % of subtotal)			1 636 000
(l) Foundation (2 % of subtotal)			156 000
(m) Testing and start up			100 000
GRAND TOTAL (a) to (m)			<u>9 984 000</u>

⁺) including machining, keys, penetrations, support etc.⁺⁺) without development cost

(hot going PCIV with cold going cables)

operation pressure, p_o MPa	1	2	4	6	10
design pressure, p_i MPa	1.8	2.8	4.8	6.8	10.8
	Cost (10^6 \$)				
(a) Cast iron, cylindrical part ¹⁾	1.260	1.260	2.160	3.060	4.860
flat plates ²⁾	.636	.793	1.038	1.236	1.558
total	1.896	2.053	3.198	4.296	6.418
(b) Cables ¹⁾	.573	.892	1.529	2.166	3.440
(c) Cable heads ¹⁾	.033	.051	.088	.124	.198
(d) + (e) Shoes including insulation	.191	.296	.508	.720	1.144
(f) + (g) + (i) Liner, Internals, etc.	.486	.486	.486	.486	.486
SUBTOTAL	3.179	3.778	5.809	7.792	11.686
(j) + (m): constant	.400	.400	.400	.400	.400
(k) + (l): 23 % of subtotal	.731	.869	1.336	1.792	2.688
GRAND TOTAL	4.310	5.047	7.545	9.984	14.774
Specific installation cost, \$/m ³	539	631	943	1248	1847

$$1) \text{ (Cost)}/(\text{cost for reference pressure}) = p_r/6.8$$

$$2) \quad \text{ " " " " " } = \sqrt{p_i/6.8}$$

TABLE XVI: COST OF REFERENCE DESIGN PCIV AS A FUNCTION OF L/D RATIO

(hot going PCIV with cold going cables)

Main data: $V = 8000 \text{ m}^3$, $p_o = 6 \text{ MPa}$, $L/D = 1 \dots 10$

L/D ratio	1	2	4	5.92	8	10
inner diameter D (m)	21.67	17.20	13.66	12.0	10.84	10.07
inner height L (m)	21.67	34.40	54.64	71.0	86.72	100.74
Cost ($10^6 \$$)						
(a) Cast iron						
cylindrical part	3.060	3.060	3.060	3.060	3.060	3.060
flat plates ¹⁾	<u>6.790</u>	<u>3.467</u>	<u>1.786</u>	<u>1.236</u>	<u>0.936</u>	<u>0.763</u>
total (a)	9.850	6.527	4.846	4.296	3.996	3.823
(b) Cables						
tangential, cylindrical part	1.302	1.302	1.302	1.302	1.302	1.302
flat plates + axial ²⁾	<u>2.622</u>	<u>1.686</u>	<u>1.095</u>	<u>0.864</u>	<u>0.723</u>	<u>0.635</u>
total (b)	3.924	2.988	2.397	2.166	2.025	1.937
(c)+(d)+(e) ³⁾	1.530	1.165	0.934	0.844	0.789	0.755
(f)+(g)+(i) ⁴⁾	<u>0.372</u>	<u>0.390</u>	<u>0.442</u>	<u>0.486</u>	<u>0.526</u>	<u>0.560</u>
SUBTOTAL	15.676	11.070	8.619	7.792	7.336	7.075
(j)+(m)	0.400	0.400	0.400	0.400	0.400	0.400
(k)+(l): 23 % of subtotal	<u>3.605</u>	<u>2.546</u>	<u>1.982</u>	<u>1.792</u>	<u>1.687</u>	<u>1.627</u>
GRAND TOTAL	19.68	14.02	11.00	9.984	9.42	9.10

$$1) (\text{cost})/(\text{cost for reference L/D ratio}) = \frac{D^3}{12^3} \frac{6.1 + L/100}{6.8}$$

$$2) \quad " \quad " \quad " \quad " \quad " \quad " = \frac{D^2}{12^2} \frac{6.1 + L/100}{6.8}$$

3) varied as (b) total

$$4) (\text{cost})/(\text{cost for reference L/D ratio}) = \frac{L + D/2}{77} \frac{D}{12}$$

TABLE XVII: COST OF REFERENCE DESIGN PCIV AS A FUNCTION OF VOLUME
(hot going PCIV with cold going cables)

Main data: $L/D = 5.92$, $p_0 = 6 \text{ MPa}$, $V = 100 \dots 10\,000 \text{ m}^3$

volume V	(m^3)	100	580	1000	8000	10000
inner height L	(m)	16.46	29.6	35.52	71	76.55
inner diameter D	(m)	2.78	5	6.0	12	12.9
		Cost ($10^6 \text{ \$}$)				
(a) Cast iron, cylindrical part ¹⁾		0.069 ²⁾	0.222	0.383	3.06	3.83
flat plate ³⁾		0.015	0.090	0.155	1.24	1.54
total		0.084	0.312	0.538	4.30	5.37
(b)+(c)+(d)+(e) ¹⁾		0.038	0.218	0.376	3.01	3.76
(f)+(g)+(h)+(i) ⁴⁾		0.026	0.084	0.122	0.49	0.56
SUBTOTAL		0.148	0.614	1.036	7.79	9.69
(i)+(l)		0.4	0.4	0.4	0.4	0.4
(j)+(h): 23 % of subtotal		0.034	0.141	0.238	1.79	2.23
GRAND TOTAL		0.582	1.155	1.674	9.98	12.32
Specific installation cost, $\text{\$/m}^3$		5820	1992	1674	1248	1232

1) $(\text{cost})/(\text{cost for reference volume}) = V/8000$

2) " " " " " = $DL/2040$

3) " " " " " = $D^3/1728$

4) " " " " " = $(L + D/2)(D/924)$

TABLE XVIII: HOT WATER OUTPUT OF EXPANSION-TYPE ACCUMULATORS

Upper operating pressure p_1	bar	10	20	40	45	60	80	100
<u>(i) Charged condition</u>								
Temperature t_1	$^{\circ}\text{C}$	180	212	250	257	276	295	311
Enthalpy saturated water, h'	kJ/kg	763	909	1087	1122	1214	1317	1408
Enthalpy saturated steam, h''	kJ/kg	2776	2797	2800	2798	2785	2760	2728
Specific volume liquid, v'	m^3/tonne	1.1274	1.1766	1.2521	1.2691	1.3187	1.3842	1.4526
" " vapor, v''	m^3/tonne	194.3	99.54	49.75	44.04	32.44	23.53	18.04
<u>(ii) Discharged condition</u>								
<u>(a) No recharge</u>								
Discharge pressure, p_2 (Fig. 5)	bar	8.6	16	30	33	42	52	64
Discharge temperature, t_2	$^{\circ}\text{C}$	173	201	234	239	253	266	280
Temperature drop, $t_1 - t_2$	$^{\circ}\text{C}$	7	11	16	18	23	29	31
Enthalpy discharged, $h'(p_2)$	kJ/kg	734	856	1008	1034	1102	1167	1236
Mean enthalpy, $h'(p_1 + p_2)/2$	kJ/kg	751	885	1050	1080	1161	1247	1327
Specific volume saturated steam, $v''(p_2)$	m^3/kg	224.3	123.7	66.63	60.53	47.31	37.82	30.23
Specific hot water output, $\rho'(p_1) - \rho''(p_2)$	tonne/m^3	0.883	0.842	0.784	0.771	0.737	0.696	0.655
<u>(b) With recharge by saturated steam</u>								
$(p_1 = p_2 = \text{constant}; t_1 = t_2 = \text{constant})$								
Recharge steam required, $\dot{m}_v = v'/v''$	-	0.006	0.012	0.025	0.029	0.041	0.059	0.081
Specific hot water output $\rho'(p_1)$	tonne/m^3	0.887	0.850	0.799	0.788	0.758	0.722	0.688
<u>(c) With live steam recharge ($p_1 = p_2 = \text{constant}; t_1 = t_2 = \text{constant}, h_{FD} = 3356 \text{ kJ/kg}$)</u>								
Ratio $\dot{m}_{vFD}/\dot{m}_v = r/(h_{FD} - h')$	-	0.776	0.772	0.755	0.750	0.733	0.708	0.678
Recharge steam required $\dot{m}_{vFD} = (\dot{m}_{vFD}/\dot{m}_v)\dot{m}_v$	-	0.005	0.009	0.019	0.022	0.030	0.042	0.055
Specific hot water output $(1 - \dot{m}_{vFD}/\dot{m}_v)\rho'(p_1)$	tonne/m^3	0.886	0.847	0.794	0.782	0.750	0.710	0.670

TABLE XIX: HOT WATER CONSUMPTION OF A FLOW SHEET WITH THREE FLASH EVAPORATORS AND A SEPARATE PEAK TURBINE

condenser pressure: 0.1 bar

Turbine inlet pressure bar			10			20			60			100		
Inlet temperature	t_i	$^{\circ}\text{C}$	180			212			276			311		
" enthalpy	h^*	kJ/kg	763			909			1214			1408		
Flash evaporators	t_i^*	$^{\circ}\text{C}$	156	137	115	180	141	115	222	169	115	246	180	115
" "	p_i	bar	5.8	3.3	1.7	10.0	4.4	1.7	24.0	7.8	1.7	37	10	1.7
" "	$h^*(p_i)$	kJ/kg	665	575	483	763	620	483	952	716	483	1065	763	483
Enthalpy drop	Δh	kJ/kg	108	90	92	146	143	137	262	236	233	343	302	280
Heat of evaporation	$r(p_i)$	kJ/kg	2089	2154	2216	2014	2122	2216	1849	2050	2216	1736	2014	2216
Steam generation ¹⁾		kg steam/kg water	0.052	0.040	0.038	0.072	0.063	0.053	0.142	0.099	0.080	0.198	0.120	0.086
Turbine enthalpy drop	Δh	kJ/kg steam	480	410	330	550	450	330	660	510	330	700	550	330
Turbine work		kJ/kg hot water	24.8	16.2	12.4	39.9	28.1	17.6	93.5	50.4	26.3	138.3	66.1	28.4
" "		$\text{kWh/tonne hot water}$	6.9	4.5	3.5	11.1	7.8	4.9	26.0	14.0	7.3	38.4	18.4	7.9
" " total		$\text{kWh/tonne hot water}$			14.9			23.8			47.3			64.7
Hot water consumption		kg/kWh			67.1			42.0			21.1			15.5

¹⁾ related to the hot water inlet flow of the first flash evaporator

TABLE XX: SPECIFIC STORAGE VOLUME AND COST OF EXPANSION-TYPE STEAM STORAGE
WITHOUT SUPERHEATING (A2)

Upper operating pressure p_1 bar	10	20	40	60	100
(a) Pressure					
Discharging pressure $p_2^{1)}$ bar	8.6	16	30	42	64
Mean pressure $p_m = (p_1 + p_2)/2$ bar	9.3	18	35	51	82
(b) Specific storage volume and storage cost					
Electric output per tonne of hot water at p_m (see Fig.57) kWh/tonne	14.2	22.2	33.8	42.2	57.4
Specific hot water rate $^{1)}$ tonnes/m ³	0.883	0.842	0.784	0.737	0.655
Specific storage capacity kWh/m ³	12.5	18.7	26.5	31.1	37.6
Specific storage volume m ³ /kWh	0.080	0.053	0.038	0.032	0.027
Specific installation cost $^{2)}$ £/m ³	539	631	943	1248	1847
Specific storage cost £/kWh	43.0	33.8	35.6	40.1	49.1

1) see Table XVIII, part (ii)(a)

2) see Table XV

TABLE XXI: SPECIFIC STORAGE VOLUME AND COST OF EXPANSION-TYPE FEED WATER AND STEAM STORAGE, WITHOUT STEAM RECHARGING (B1)

Upper operating pressure p_1	60			80			100		
(a) Basic data of base load plant									
Electric output MW							650		
Feed water temperature °C							250		
Feed water enthalpy kJ/kg							1087		
Feed water flow rate tonne/hr.	approx.						1980		
Reheater inlet pressure bar							41		
Live steam enthalpy kJ/kg							3356		
(b) Steam rate									
Pressure beginning, mean, bar end	60	51	42	80	66	52	100	82	64
Excess enthalpy Δh kJ/kg	127	74	15	230	160	80	321	240	149
Steam generation $x = \Delta h / r_{40} = \Delta h / 1713$	.074	.043	.009	.134	.093	.04	.187	.140	.087
Hot water from storage tonne/hr. $m_s = 1980 / (1-x)$	2138	2069	1997	2287	2184	2077	2437	2303	2169
Additional reheat steam tonne/hr.	159	89	17	307	204	97	456	323	189
(c) Peak power									
LP-Power increase MW	31.4	17.6	3.4	60.5	40.2	19.1	89.9	63.7	37.3
Power increase by feed water train disconnection MW	-	105	-	-	105	-	-	105	-
Additional peak power MW	136.4	122.6	108.4	165.5	145.2	124.1	194.9	168.7	142.3
Additional peak power %	20.7	18.6	16.4	25.1	22.0	18.8	29.5	25.6	21.6
(d) Storage volume									
Specific electric power, kWh/t K_{sp}/h_s	59.3			66.5			73.3		
Specific hot water flow tonne/hr. rate (Tab. XVIII, if a)	.737			.696			.655		
Specific storage capacity kWh/m ³	43.7			46.3			48.0		
Specific storage volume m ³ /MWh	22.9			21.6			20.8		
(e) Storage cost (PCIV only)									
Specific storage cost \$/m ³	1248			1548 ⁺⁾			1847		
Specific storage cost \$/kWh	28.6			33.4			38.4		

⁺⁾ Interpolated, otherwise from Table XV

TABLE XXII: SPECIFIC STORAGE VOLUME OF EXPANSION-TYPE FEEDWATER AND STEAM STORAGE, WITH LIVE STEAM RECHARGING (B2)

Operating pressure	P_0	bar	41 ^{*)}	45	60	80	100
(a) Basic data of base load plant							
Electric output	MW					650	
Feed water temperature	$^{\circ}\text{C}$					250	
Feed water enthalpy	kJ/kg					1087	
Feed water flow rate	tonne/hr.				approx.	1980	
Reheater inlet pressure	bar					41	
Live steam enthalpy	kJ/kg					3356	
(b) Steam rate							
Excess enthalpy to feed water $\Delta h = h' - h_{sp}$	kJ/kg		0	35	127	230	321
Steam generation $x = \Delta h / r_{40} = \Delta h / 1713$	-		0	.020	.074	.134	.187
Hot water from storage $\dot{m}_s = 1980 / (1 - x)$	tonnes/h		1980	2021	2138	2287	2437
Additional reheat steam $\Delta \dot{m}_{2U}$	tonnes/h		0	41	159	307	456
Live steam recharge $\Delta \dot{m}_{FD} = 1980(\dot{m}_{vFD})^{+++}$	tonnes/h		0	44	59	83	109
(c) Peak power							
LP-Power increase $\Delta \dot{m}_{20}(710/3600)$	MW		-	80	31.4	60.5	89.8
MP-power increase $\Delta \dot{m}_{FD}(1090/3600)$	MW		-	13.3	17.9	25.1	33.0
Difference			-	5.3	13.5	35.4	56.8
Power increase by feed water train disconnection	MW		105	105	105	105	105
Additional peak load N_{Sp}	MW		105	99.7	118.5	140.4	161.8
" " "	%		15.9	15.1	18.0	21.3	24.5
(d) Storage volume (PCIV)							
Specific electric power $N_{Sp}/\dot{m}_s$	kWh/tonne		53.0	49.3	55.4	61.4	66.4
Specific hot water flow (Table XVIII, part ii, c)			0.794	0.782	0.750	0.710	0.67
Specific storage capacity	kWh/m ³		42.1	38.6	41.6	43.6	44.5
Storage volume	m ³ /MWh		23.8	25.9	24.0	22.9	22.5
(e) Storage cost (PCIV only)							
Specific storage cost	\$/m ³		958 ⁺⁺⁾	1020 ⁺⁺⁾	1248	1548 ⁺⁺⁾	1847
Specific storage cost	\$/kWh		22.8	26.4	30.0	35.5	41.6

^{*)} pure feed water storage ⁺⁺⁾ interpolated, otherwise from Table XV

⁺⁺⁺⁾ see Table XVIII, part (ii, c)

Fig. 1 Possibilities of load balancing /46/

Fig. 2 Basis possibilities of thermal energy storage /23/

A, B Storage medium of constant specific energy
(A: solid, B: liquid)

C, D Change of specific energy
(C: solid, D: liquid)

----- Charging

———— Discharging

Fig. 3 Main types of hot water accumulators /23/

A: Displacement storage

B: Expansion-type storage

C, D: Varying pressure storage (internal steam generation)

-----> Charging

————> Discharging

Fig. 4 Specific storage capacity of a varying pressure accumulator /49/

Example:

Upper pressure: $p_1 = 60$ bar, lower pressure: $p_2 = 20$ bar,
 initial water volume: 85 %, initial steam volume (steam cushion): 15 %

Specific storage capacities: 119 kg steam per 1 m³ initial water volume
 19 kg steam per 1 m³ initial steam volume

Specific storage capacity of accumulator:

$$(0.85)119 + (0.15)19 = 104 \text{ kg steam per 1 m}^3 \text{ accumulator volume}$$

Fig. 5 Specific storage capacity of an expansion-type accumulator

Fig. 6 Possibilities of pressurized air storage for gas turbine plants

A: Displacement air storage

B: Varying pressure air storage, underground

C: Varying pressure air storage, above ground

Fig. 7 Discharge diagram for air storage /11/

For instance:

pressure ratio	P_2/P_1	= 0.06
heat capacity ratio	W_{Fe}/W_o	= 5
temperature ratio	T_2/T_1	= 0.94
density ratio	ρ_2/ρ_1	= 0.064

Fig. 8 Specific storage capacity (kWh of electric energy per m^3 storage volume)

A Hydraulic pumped storage

B1 Gas turbine with varying pressure air storage
($p_1/p_2 = 1.5$)

B2 Gas turbine with displacement-type air storage

C1 Varying pressure steam storage

C2 Feed water storage

Fig. 9 Riveted steam accumulator vessel of the Berlin-Charlottenburg plant /51/

Fig. 10 The Berlin-Charlottenburg steam storage plant (photo: G. Beckmann, 1972)

Fig. 11 Flow sheet of a steam storage installation in a BOF plant /93/

Fig. 12 Steam accumulator for a US steel plant during erection (photo: American Waagner-Biro, Inc.)

Fig. 13 Correlation between compressive and tensile strength of grey cast iron types /78/

Fig. 14 Smith diagram for grey cast iron, i.e. alternating stress vs. means stress /103/

Fig. 15 Multi-axial stresses and stress limits for cast iron and steel in the Mohr/Leon plot, i.e. shear stress τ vs. direct stress σ

Fig. 16 Interaction between prestressing system and cylindrical part for prestressed pressure vessels

Fig. 17 Interaction between prestressing system and upper flat plate for prestressed pressure vessels

Fig. 18 Siempelkamp design of a PCIV for the 300 MW-THTR (Thorium High Temperature Reactor) being built by HRB at Schme-hausen, FRG /84/

Fig. 19 PCIV test vessel for 9 MPa pressure, built by Siempelkamp in 1972 /84/

Fig. 20 Siempelkamp design of PCIV
for a 1450 MW Boiling Water
Reactor /84/

Fig. 21 Details of liner and internal thermal insulation of a PCIV according to Fig. 20 /84/

- 1 Circumferential wires
- 2 Vertical tendon
- 3 Cast iron block
- 4 Liner
- 5 Water gap (65°C)
- 6 Cooling system
- 7 Outer insulation casing
- 8 Thermal insulation (metallic foil or wire mesh)
- 9 Inner insulation casing

Fig. 22 PCIV for 23 MPa helium control
rod drive of the THTR Schmehausen
/11/
(Ø means DIA)

Fig. 23 Stress-Strain diagram for a cold-going PCIV

Fig. 24 Stress-strain diagram for a hot going PCIV with cold going cables (conservative design base)

Fig. 25 Stress-strain diagram for a hot going PCIV with cold going cables (advanced design base)

Fig. 26 Permissible steady-state operation ranges for a cold going PCIV

Fig. 27 Permissible steady-state operation ranges for a hot going PCIV with cold cables (conservative design base)

Fig. 28 Permissible steady-state operation ranges for a hot going PCIV with cold cables (advanced design base)

Fig. 29 Reference design, general arrangement

- 1 Cylindrical wall
- 2 Flat plate
- 3 Liner
- 4 Tendons
- 5 Charging/discharging pipes
- 6 Supports
- 7 Structural steel
- 8 Safety valve
- 9 Trolley

Fig. 30 Alternative reference design

- 1 Cylindrical wall
- 2 Flat plate
- 3 Liner
- 4 Tendons
- 5 Charging/discharging pipes
- 6 Supports
- 7 Structural steel
- 8 Safety valve
- 9 Trolley

SECTION A-A

Fig. 31 Cross-section of
alternative reference design
(Legend as for Fig. 30)

Fig. 32 Details of upper and lower flat plate

1 Cast iron blocks	6 Shoe
2 Keys	7 Tangential tendons
3 Cylindrical wall	8 Axial tendons
4 Liner	9 Pressurized insulation
5 Discharge pipe	10 Insulation

Fig. 33 Heads of axial cables

Fig. 34 Details of tangential cable arrangement

Fig. 35 Details of shoes for tangential cables, cylindrical part

1 Shoe	5 Cylindrical wall (cast iron)
2 Tangential tendons	6 Key
3 Wire catcher	7 Liner
4 Pressurized insulation	

Fig. 36 Dimensionless thermal stress response of the surface of a cylindrical shell to a thermal ramp function, vs. Fourier Number t^+

(a) vs. Fourier Number t^+

(b) vs. reciprocal Biot Number $1/\alpha^+$

Fig. 37 Dimensionless thermal stress response of the surface of a cylindrical shell to a thermal shock

Fig. 38 Thermal stresses in a displacement-type storage vessel

Fig. 39 Permissible thermal stresses, operational conditions

Fig. 40 Permissible temperature variations in cylindrical part and flat plate

Fig. 41 Girder-type flat plate

- 1 Cast iron blocks
- 2 Keys
- 3 Cylindrical wall

Fig. 42 Flow sheet suitable for the use of welded steel storage vessels, employing varying pressure storage, sequential discharge, separate peak load turbine and multi-pressure condenser /33, 37, 39/

1 Reactor (PMR)	8 Superheat accumulator
2 Steam Generator	9 Superheater
3 Main (base load) turbine	10 Discharge steam lines
4 Main condenser	11 Peak load turbine
5 Feed heater train	12 Peak load condenser
6 Charge line	13 Cooling tower
7 Main accumulators	14 Condensate storage

Fig. 43 Expansion diagram for main turbine (LWR) and peak-load turbine according to the flow sheet Fig. 42 /33, 37, 39/

- Main turbine
- Peak load turbine

Fig. 44 Flow sheet suitable for the use of a PCIV employing varying pressure type storage and separate peak-load turbine /42/

- 1 Varying pressure accumulator
- 2 Charge steam line
- 3 Discharge steam line
- 4 Superheat accumulator
- 5 Superheater
- 6 Peak load turbine
- 7 Condenser
- 8 Condensate storage

Fig. 45 Flow sheet suitable for the use of welded steel storage vessel, employing displacement-type storage, external steam generation and separate peak-load turbine /37, 39/

1 Reactor (HTR)	9 Superheater
2 Steam Generator	10 Discharge steam lines
3 Main (base load) turbine	11 Peak load turbine
4 Main condenser	12 Peak load condenser
5 Feed heater train	13 Helium blower
6 Charge line	14 Condensate storage
7 Main accumulators	15 Jet pump
8 Superheat accumulator	16 Start-up valve

Fig. 46 Flow sheet suitable for the use of PCIV, employing expansion-type steam storage with superheat storage vessel, and a separate peak-load turbine /42/

- 1 Expansion-type accumulator
- 2 Water discharge line
- 3 Flash evaporators
- 4 Cold storage
- 5 Superheater
- 6 Steam lines
- 7 Peak-load turbine
- 8 Condenser
- 9 Condensate storage
- 10 Superheat accumulator

Fig. 47 Flow sheet suitable for the use of PCIV, employing expansion-type steam storage, with superheating by means of branched-off stored hot water, and a separate peak-load turbine /42/

- 1 Expansion-type accumulator
- 2 Water discharge line
- 3 Flash evaporators
- 4 Cold storage
- 5 Superheater
- 6 Steam lines
- 7 Peak-load turbine
- 8 Condenser
- 9 Condensate storage

Fig. 48 Flow sheet according to Fig. 47, with thermal and flow data

1 Heat source	7 Feed water tank
(cold storage)	11 Peak load tur-
bine, LP
2 Expansion-type PCIV	8 Feed water heater	12 Peak load tur-
bine, condenser
3 Main turbine, HP	9 Flash evaporator	
4 " " , IP	10 Peak load turbine, HP	
5 " " , LP		
6 Main condenser		

Fig. 49 Expansion-type feed water and steam storage with separate peak load turbine

Fig. 50 Expansion-type feed water and steam storage with overloadable main turbine (legend of Fig. 49 refers).

Fig. 51 Flow sheet similar to Fig. 50, but with HP mixing preheater (13)
(legend of Fig. 49 refers)

Fig. 52 Solar power station with PCIV varying pressure-type storage for instantaneous reserve, acting alternatively or simultaneously as expansion-type feed water storage /46/

- 1 Solar steam generator
- 2 HP turbine
- 3 LP turbine
- 4 PCIV accumulator
- 5 Mixing condenser
- 6 Dry cooling tower
- 7 Feed water heater

Fig. 53 Solar power station with expansion-type storage in a combined feed water and steam storage arrangement

- 1 Solar steam generator
- 2 Turbine
- 3 PCIV accumulator
- 4 Superheat accumulator
- 5 Cooling tower
- 6 Water discharge line
- 7 Flash evaporators
- 8 Superheater
- 9 Feed water heaters
- 10 Mixing condenser

Fig. 54 Cost of PCIV as a function of operating pressure p_0

Fig. 55 Cost of PCIV as a function of L/D ratio

Fig. 56 Specific cost of PCIV as a function of operating pressure p_0 , for various volumes

Fig. 57 Electric energy gained from 1 tonne of hot water vs. upper operating pressure p_1 in a flow sheet as shown in Fig. 47, for saturated and superheated steam

Fig. 58 Specific installation cost C_s of PCIV for TES, vs. operating pressure p_o

- (A) Expansion-type steam storage, with multi-state flash evaporating (Fig. 46, 47, 48)
 - (A1) with superheating (20 and 50 K, respectively)
 - (A2) without superheating
- (B) Expansion-type feedwater and steam storage (Fig. 49, 50, 51, 53)
 - (B1) without recharge
 - (B2) with live steam recharge

Fig. 59 Installation cost C of a peak-load plant with expansion-type PCIV steam storage (A), and with a separate peak-load turbine (Fig. 47) as a function of operating pressure p_o

Fig. 60 Installation cost C of a peak-load plant with combined feed water and steam storage (B), and a separate peak-load turbine (Fig. 49), as a function of operating pressure p_0

Fig. 61 Installation cost C of a peak-load plant with expansion-type feedwater and steam storage (B), with overloadable main turbine (Fig. 50, 51), as a function of operating pressure p_o

Fig. 62 Peak power in percent of base load power for expansion-type combined feed water and steam storage (B), with either separate peak-load turbine (Fig. 49) or overloadable main turbine (Figs. 50, 51) as a function of operating pressure p_o

Fig. 63 Installation cost C of an expansion-type steam storage plant (A), with separate peak-load turbine (Figs. 46, 47, 48), vs. condenser back pressure p_c

Fig. 64 Installation cost C of an expansion-type steam storage plant (A), with separate peak-load turbine (Figs. 46, 47, 48), vs. degree of superheating Δt

Fig. 65 Modes of operation and lay-out of a solar power plant with TES

Fig. 66 Initial cost of turbine and thermal energy storage of a solar power plant with TES according to Fig. 65, vs. turbine lay-out factor ϕ

Fig. 67 Off-peak power related portion c_3 of peak power cost of a TES plant, as a function of off-peak power cost (fuel cost) c_{off} and storage efficiency η_{st}

Fig. 68 Fixed cost c_{12} of a TES plant

Fig. 69 Quick start-up of a gas turbine plant with PCIV air storage /11/

- 1 Start-up damper
- 2 Non-return valve
- 3 Vent
- 4 Air storage PCIV
- 5 Throttling valve
- 6 Charging compressor

Fig. 70 Start up procedure of a gas turbine plant with PCIV air storage /11/

Fig. 71 Required volume of start-up air storage for a gas turbine plant as a function of start-up time required and storage pressure