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**ALTERNATE APPLICATIONS OF FUSION POWER—
DEVELOPMENT OF A HIGH-TEMPERATURE BLANKET
FOR SYNTHETIC-FUEL PRODUCTION**

by

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Fusion Power Program

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ABSTRACT

This study has shown that utilization of the unique features of a fusion reactor can result in a novel and potentially economical method of decomposing steam into hydrogen and oxygen. Most of the power of fusion reactors is in the form of energetic neutrons. If this power could be used to produce high temperature uncontaminated steam, a large fraction of the energy needed to decompose the steam could be supplied as thermal energy by the fusion reaction. Proposed high temperature electrolysis processes require steam temperature in excess of 1000°C for high efficiency. The design put forth in this study details a system that can accomplish that end.

A 3054 MW gross STARFIRE type reactor was assumed to be available and the high temperature beds were designed to be compatible with that machine. Steam, at a pressure of 1.7 atm and at a temperature of 1000°C, is produced at a rate of 409 kg/s. This represents 1882 MW of power. Most of the power (1045 MW) needed to produce this steam comes from a low temperature (121°C) steam generator. Breeding blanket coolant could easily supply this power. Approximately 271 MW of steam at 460°C (typical of what might be first-wall coolant) is also needed. The balance of the power (566 MW) required is from neutron capture by the pellets at high temperature (1100°C).

Section 1

INTRODUCTION AND SUMMARY

1. INTRODUCTION AND SUMMARY

Fusion has been considered primarily as a source for electric power generation. However, since ~80% of the U.S. energy consumption is nonelectrical, it is desirable to consider fusion for nonelectrical applications. The primary goal of the present study is to design a fusion reactor blanket which is capable of producing high temperature process heat which can be used in nonelectrical applications such as the production of synfuels. The study is a continuation of earlier work that showed the general feasibility of a high temperature blanket for producing pressurized steam at temperatures in excess of 1000°C.¹⁻⁴ The goals of the study during FY 1981 were to complete and optimize the blanket and heat exchanger design, examine the effects of off-normal conditions, and examine the effects of the high temperature radiation environment on the blanket materials.

1.1 Design Concept

The fusion reactor design used for this study is based upon previous reactor design studies. The major modifications made to this 7-m major radius design were to increase κ , the plasma elongation, in order to maximize the vertical blanket geometry, and to increase the size of the TF coils to accommodate the upper and lower shielding. The reference reactor design parameters are presented in Table 1-1 and the reactor cross section is shown in Fig. 1-1. Note that the reference burn cycle is 30 min with a 1-min downtime. More recent design studies such as STARFIRE⁵ have shown the feasibility of steady-state operation. No attempt to upgrade the reference fusion driver was made in this study, however. The major emphasis has been placed on the high temperature blanket design. The tritium breeder material is Li_2O which is placed immediately behind the first wall. Details of the blanket geometry are presented in Sec. 3 along with the thermal hydraulic calculations.

The concept selected for the production of high temperature steam is shown schematically in Fig. 1-2. In this concept the neutrons produced by the fusion process deposit their energy in the blanket which consists of ceramic pebbles. The pebbles are continuously circulated to extract the heat from the blanket.

Table 1-1. Design Parameters of Reference Reactor

Major radius	7.0 m
Minor radius	2.3 m
Aspect ratio	3.0
Elongation factor (κ)	1.68
Vacuum scrape-off region	0.2 m
Plasma volume	950 m ³
Wall area	1143 m ²
TF field (maximum at magnet)	9 T
Beta (toroidal)	0.07
Beta (poloidal)	1.75
Safety factor	3.0
Plasma current	11.6 MA
Average electron temperature	8 keV
Average ion temperature	7.95 keV
Average ion density	$1.5 \times 10^{20} \text{ m}^{-3}$
Confinement parameter ($\text{m}\tau$)	$3.2 \times 10^{20} \text{ s-m}^{-3}$
Average alpha particle density	$1.83 \times 10^{20} \text{ m}^{-3}$
Peak thermal power	
Alpha energy	469 MW
Neutron energy	2584 MW
Burn time	30 min
Down time	1 min
Neutron wall loading (peak average)	1.67 MW/m ²
Total wall loading	2.08

Fig. 1-1. High temperature falling bed reactor reference.

Fig. 1-2. Schematic representation of the pebble bed blanket system.

The pebbles then pass through a heat exchanger where they transfer their heat to a gas stream. The cooled pebbles are finally returned to the reactor via a preheater where they collect some of the lower temperature heat generated in the first wall and tritium breeding blanket. This concept was proposed because it avoids the temperature limitation (and temperature drop) associated with a conventional heat exchanger which uses a barrier between two fluid streams. In addition, since damaged pebbles can be easily replaced, the falling bed design is expected to be less sensitive to material damage than a static design.

The pellets form a falling packed bed as they move through the duct. This duct is vertical, straight, and fabricated of Type 316 stainless steel. The assembly of ducts would form a cylinder around the torus. In the present design, 36 ducts would completely encompass the torus. Another 36 ducts not illustrated in the Fig. 1-2 would be located in the inboard region of the blanket. Of the 3054 MW produced, approximately 500 MW is deposited directly on to the first wall, 700 MW is absorbed by the pellets and the duct wall, and the balance is absorbed within the tritium breeding blanket volume. Most of the neutrons are not ultimately captured by the pellets and hence are available for power production and tritium breeding.

The present design is based upon a fusion reactor which has 12 toroidal field coils. A plan view of part of the reactor torus is shown in Fig. 1-3. The design has 36 inner ducts, 3 of which are shown. Pellets would be supplied to three inner and three outer ducts by one of twelve preheaters. In a similar manner, the hot pellets discharged from these ducts would be directed to one of the twelve heat exchangers. A simplified view of the pellet heat transfer system is shown in Fig. 1-4. A steam generator needed to produce the makeup steam is included. It can be seen that this is the largest power consuming component. However, because its temperature is so low, supplying this large number of megawatts should not be a problem. If rejected steam from the synfuel plant was returned and the nondecomposed steam was used in a steam generator to replace makeup steam, then the present steam generator would be eliminated (see Sec. 3.5). Overall parameters, temperatures, and flow rates for the high temperature system are given in Table 1-2.

As shown in Fig. 1-3, the inner and outer ducts would completely enclose the torus. Such an arrangement would prevent access to the torus of essential

Fig. 1-3. Reactor plan view.

equipment (such as vacuum pumps or neutral beam injectors). The present design is a simplification of what would be proposed as a working arrangement. This simplification is based upon neutronic calculations. In a working arrangement, access to the torus would be done by relocating some portions of the falling beds to other regions. The volume of the ducts, pellets, steam, etc., would remain constant; however, their location would change.

All of the inner and outer ducts are of a relatively simple design. There are 36 inner and 36 outer ducts with an active (neutron capture) heating length of 10 m. The walls are hollow and force-cooled with steam. All inner surfaces are lined with Feltmetal which acts as a passive thermal shield. Figures 1-5 and 1-6 illustrate the ducts. Notice that the outer duct is physically larger, has a larger inventory of pellets and has internal walls which act as braces. These braces are identical in design to the other walls and hence are force cooled. Their purpose is to add strength to the outer duct and reduce the stress within and the bulging of the long (1.705 m) walls.

Fig. 1-4. Falling bed design (cross section). Overall height is 43 m.

Fig. 1-5. Inner duct configuration.

Fig. 1-6. Outer duct configuration.

Table 1-2. High Temperature System Parameters

Gross fusion reactor power	3054 MW
Total process steam produced	1882 MW
Production rate	409 kg/s
Steam pressure	1.7 atm
Steam temperature	1100°C
Heat transferred to steam via net neutron capture	566 MW
Heat transferred to pellets via preheater steam	271 MW
Makeup process steam needed	1045 MW @ 121°C
409 kg/s @ 2 atm	
Duct wall steam coolant needed	212 kg/s @ 4 atm
145°C	
Steam rejected by duct walls	3.2 atms
443°C	
Steam needed for preheaters	409 kg/s
1.5 atm	
460°C	
Steam rejected by preheaters	1.3 atm
142°C	
Total inventory of pellets	2.9×10^6 kg
Electric power needed for pellet transport system assuming 75% efficiency	344 kW

Pellets enter the duct at the top and are assumed to be isothermal (450°C). As the pellets move down the duct a radial temperature gradient will develop within the duct as a consequence of uneven heating. Pellets farther from the plasma source are shadowed by those ahead and have a lower heat generation rate. This is most severe in the outer duct which has 36 cm of pellets. The heating rate varies from 5.8 W/cm³ to 0.81 W/cm³. Those pellets closest to the plasma heat the quickest. In an effort to avoid too high a temperature, the exit of the duct is deliberately shifted to the inboard side so that the hottest pellets have the greatest velocity. It was not known how this skewed exit would influence the velocity profile upstream of the exit and hence experiments were conducted (see Appendix A) to determine the best angle for inducing a skewed velocity profile. Speeding up the pellets closest to the plasma and slowing down the pellets farthest from the plasma would allow a higher bulk pellet temperature.

The duct walls require active steam cooling. In the unlikely event of a loss of coolant, the duct wall would overheat rapidly. This necessitated the inclusion of a release mechanism to allow the rapid dumping of the pellets within the duct. Dumped pellets could be accommodated in the heat exchangers.

1.2 Potential High Temperature Electrolysis Processes

The preliminary feasibility study included an assessment of the falling bed blanket as a source of thermal energy to be used in the production of hydrogen by electrolysis.¹ It was found that the proposed characteristics of the blanket heat output (temperature in excess of 1000°C and pressures of the order of 2 atm) result in good electrolyzer performance, with a hydrogen production efficiency of 51%. In addition, several alternate thermochemical cycles have been identified that can be successfully powered by the present heat generation system; these include the sulfur hybrid cycle, the sulfur/iodine process, and the tin oxide process. The alternate thermochemical cycles are briefly discussed below.⁶⁻⁸

The hybrid/sulfuric acid cycle involves an electrolysis stage:

followed by vaporization of the H_2SO_4 , which is then decomposed at high temperature in order to regenerate the sulfur dioxide:

The resulting oxygen is vented to the atmosphere after separation of the water and sulfur dioxide by either compression and cooling or gas absorption using activated charcoal. Early studies gave efficiencies of ~44% for this cycle, whereas Ref. 6 shows the gain in efficiency (up to 54%) that results from the use of a high temperature reactor.

The sulfur/iodine process involves a low-temperature reaction in aqueous solution:

followed by thermal cracking and consideration of the iodine:

and regeneration of the SO_2

The decomposition of H_2SO_4 , compared to both sulfur-based processes, is a highly endothermic reaction, consuming most of the power required by the cycles. Typically, it is assumed to take place at 870°C , the maximum feasible temperature for an HTGR with an intermediate coolant loop. At this temperature (and at low pressures) only about 63% of the acid is decomposed per pass; this efficiency like that of the preceding cycle can be significantly increased by using a heat source at the 1100°C postulated for the falling-bed blanket. There is, furthermore, an additional advantage to the use of the blanket design proposed here (in comparison with other fusion blankets), and this has to do with the nature of the decomposing sulfur dioxide. At the high temperatures that are desirable in order to achieve a satisfactory efficiency, alumina and magnesia are some of the few materials that can withstand the highly corrosive environment. The possibility of circulating the reacting gases through a fusion blanket has been considered, except that neutron activation is seen as a potential problem; again, the heat generation system proposed here, with the cooling gas passing only through a heat exchanger placed outside the neutron shielding, appears to be a particularly suitable arrangement.

1.3 Summary of Results

A detailed study of the major falling pebble bed components has been conducted to determine the engineering viability of such a system. The study included a materials analysis, thermal hydraulic analysis, and stress analysis. In addition, an experiment has been conducted to determine the flow characteristics of the pebble bed.

The materials analysis, Sec. 2, concentrates on the selection of materials for the ducts within the blanket. Previous work on the high temperature

blanket has identified the interface between the hot ($\sim 1100^{\circ}\text{C}$) pellets and the cool ($\sim 500^{\circ}\text{C}$) duct walls as a critical materials problem area.² Previously this interface consisted of a thick (~ 1 cm) ceramic liner. However, initial analysis of the liner indicated that the liner could not withstand high stresses and would be subject to unacceptable radiation damage. Therefore, an alternate interface structure was selected. The concept finally selected is known commercially as Feltmetal,⁽⁷⁾ which consists of low density sintered metal fibers. A number of metals, including iron, nickel, cobalt, and copper alloys, have been used to make Feltmetal structures. Their principal properties are low thermal conductivities and low elastic moduli which results in high temperature differences with low thermal stresses in high heat load applications. Feltmetal structures can be fabricated and joined using standard techniques. These materials have been successfully used in many high temperature operations where metal and ceramic structures are coupled together. The potential problems for Feltmetal in the high temperature pebble duct include long term effects of corrosion and radiation damage, and a developmental program is required to assess these problems and to determine a working lifetime. In the absence of experimental data for the duct operating conditions, the maximum operating temperature of the Feltmetal has been set at 1000°C , which is within the range where it has been used for other applications.

The results of the thermal hydraulic analysis, Sec. 3, indicate that a total of 1882 MW of 1100°C steam can be produced at a rate of 409 kg/s and at a pressure of 1.7 atm. Most of the power (1045 MW) needed to produce the steam comes from a low temperature (121°C) steam generator which can be heated by the breeding blanket coolant. Approximately 271 MW of steam at 460°C from the first wall is also used. The balance of the power (566 MW) comes from neutron capture within the falling pebble bed. This system is a once through system where the high temperature steam is supplied to an outside user and not recirculated. If the steam were recirculated back to the reactor, then the low temperature steam generator and preheaters would be eliminated from the system. The recirculating system would, therefore, reduce the number of components, reduce the pellet inventory, and reduce the size of the transport system. The total amount of fusion power used for steam heating would be 566 MW. The rest of the fusion power would be available for electric power generation.

The heat exchanger, preheater, and inner and outer ducts were examined structurally (Sec. 4). The components were assumed to operate at $\sim 500^{\circ}\text{C}$ and were constructed from annealed Type 316 stainless steel. Forces due to internal gas pressure, pellet pressure, dead weight, and temperature differences were considered. The resultant stresses were calculated and then compared to allowable stress intensities. In all cases, the structures met the requirements of the ASME Boiler and Pressure Vessel Code. In the case of the heat exchanger, the required wall thicknesses are 2.5 cm for the top and the sides, and 2.75 cm for the bottom. The preheater, which is smaller than the heat exchanger, has required wall thicknesses of only 0.8 cm for the top and the sides, and 2.0 cm for the bottom. A 2.0-cm thick corrugated panel was used to construct the rectangular blanket ducts. This structure met the stress requirements for both normal and off-normal conditions. Buckling and natural vibrations are shown to be of little concern.

Finally, an experiment was conducted to examine the flow characteristics of the pebbles. The flow rate was determined by measuring the attenuation of a gamma-ray source through the pellets flowing out of the duct. The test results were input directly into a computer, and a computer program was written to analyze the raw data. The results are presented in Appendix A.

Section 2

MATERIALS CONSIDERATIONS

2. MATERIALS CONSIDERATIONS

2.1 Pebble Materials

Previous analysis of pebble materials have examined the areas of neutronics, induced activity, sintering, and radiation damage.^{1,2} The materials that have been examined are Al_2O_3 , MgO , SiO_2 , and $\text{MgO}\cdot\text{Al}_2\text{O}_3$. SiO_2 was eliminated from consideration because the amount of energy deposited in the high temperature region is the lowest of the materials considered. The other three materials have sufficiently short lived activation products to make them suitable as pebble materials, with MgO exhibiting the lowest activation. An analysis of the sintering potential of the pebbles at elevated temperatures indicates that sintering should not occur at temperatures $<1500^\circ\text{C}$ for a pellet diameter of 1 cm. There is little data available on the radiation damage effects in ceramic materials. The available information indicates that MgO and $\text{MgO}\cdot\text{Al}_2\text{O}_3$ are favored because of their cubic crystal structures. Radiation-induced dimensional changes are isotropic for these materials and therefore little or no microcracking is observed. Al_2O_3 exhibits anisotropic dimensional changes and microcracking during irradiation. Based upon these considerations MgO has been selected as the pebble material with $\text{MgO}\cdot\text{Al}_2\text{O}_3$ as the backup material.

2.2 Duct Material

The structural material chosen for the ducts is Type 316 stainless steel. This material has been extensively reviewed elsewhere,^{5,9} and a detailed discussion will not be presented here. Type 316 stainless steel is easily fabricated and there is a large radiation data base. The maximum operating temperature is set by the onset of helium embrittlement and the loss of tensile strength. For the present design, that temperature has been set at 500°C . The mechanical properties used for the stress analysis of Sec. 4 are those of annealed material. The radiation damage will increase the tensile strength, so that the duct stress analysis is expected to be conservative.

2.3 Interface Between Pebbles and Structure

A crucial area for the design of the high temperature blanket is the interface between the hot pebbles and the cool stainless steel duct. An earlier design of the duct incorporated a ceramic liner between the pebbles and wall.² The liner was 1 cm thick, and it was spaced 1.25 cm from the wall. Several serious problems were identified with the ceramic liner which led to a design modification. First, ceramic materials generally have low thermal conductivities, which results in large thermal gradients and thermal stresses through the liner thickness. Second, ceramic materials are generally weak in tension or bending, and thus there is a concern whether the liner could withstand the stresses due to the weight of the pellets. Finally, most ceramics have a low radiation damage tolerance compared to the structural metals being considered for fusion applications. The limited lifetime means that the ceramic liner would have to be replaced more frequently than the stainless steel duct.

The concept chosen to replace the ceramic liner is known commercially as "Feltmetal".¹⁰ Feltmetal is a porous metal product made by sintering aggregates of metal fibers. A large variety of metal alloys at different densities are available commercially. The alloys which have been used include austenitic and ferritic stainless steels, nickel alloys, cobalt alloys, and copper alloys. The density of the Feltmetal ranges from 10% to 70% of the theoretical density, with 10-20% density being the most common. Generally, these products can be fabricated using standard techniques such as rolling, pressing, punching, machining, welding, and brazing. The major advantage of these structures is that the low density results in low values of the thermal conductivity combined with low values for the elastic modulus. This combination results in low stresses even though there is a high thermal gradient through the material. In addition, since the structure can be composed of a radiation resistant metal, the Feltmetal should have a longer reactor lifetime compared to a ceramic liner. Typical properties for 35% dense material are shown in Table 2-1. It should be noted that the thermal conductivity values can be easily tailored by changing the density or the thickness.

The main concern with the use of Feltmetal is its corrosion resistance in the high temperature steam environment. These products have been used at high temperatures, and FeCrAlY and FeCrAlSi alloys have been found to offer the

Table 2-1. Typical Properties of Feltmetal

	Test Temperature (°C)	
	21	815
Mechanical Properties at 35% dense^a		
<u>Property and Test Plane^b</u>		
Ultimate strength (mPa)		
Tensile		
X-Y plane	33.4	6.9
Z plane	6.9	5.2
Shear	15.2	4.1
0.2% Offset yield strength (mPa)		
Tensile		
X-Y plane	31.7	5.9
Z plane	5.8	4.8
Shear	13.4	3.8
Compressive, Z plane	22.7	3.6
Strain at UTS (%)		
Tensile		
X-Y plane	1.0	5.6
Z plane	7.5	11.0
Shear	13.5	14.0
Modulus of elasticity (GPa)		
Tensile		
X-Y plane	7.9	1.8
Z plane	0.17	0.06
Shear	0.15	0.09
Compressive, Z-plane	0.82	0.18
Flexure	8.3	-
<u>Thermal Conductivity^c (W/m²K)</u>		
35% density	48.3	120.4
41% density	73.8	204.5

^aProperties will be 40-50% higher at 40% dense.

^bX-Y plane: property parallel to sheet dimensions.

Z plane: property parallel to thickness dimension.

^cThermal conductivity through the thickness of "Z" plane for 1/16-in. thick pad. Thicker materials will have a lower thermal conductivity.

best oxidation/hot corrosion resistance of the alloys tested. One method which is presently employed to reduce corrosion is to plasma spray the exposed Feltmetal surface with a thin ceramic layer. The ceramic provides a protective layer, but it is not required to provide structural support. Therefore, the plasma sprayed layer may crack without serious consequences, unlike the thick ceramic liner. The use of plasma spraying is attractive in this application because in-situ repair of the duct wall surface is possible.

There are also uncertainties in the use of Feltmetal which require additional development. In particular, the corrosion rates of various metals with and without ceramic coatings should be investigated for the proposed duct environment. The bonding technique of the Feltmetal to the duct should be investigated. A welding procedure is probably preferred, since no additional material is introduced into the system. If a braze is required, then the braze properties need to be examined in order to estimate its lifetime. Finally, the radiation lifetime should be examined. In the absence of appropriate data, the allowable temperature limit of the Feltmetal has been tentatively set at 1000°C, which is within the range where these materials have been used. A significant development program is required to find the best materials and fabrication procedures and to determine the maximum operating temperature for the Feltmetal.

3. THERMAL HYDRAULICS

The thermal analysis of the flowing pebble bed is presented. The objective of the analysis was to devise a system which supplies 1100°C steam to an outside user for such purposes as water decomposition or synfuel production. It has been assumed that the steam is not recirculated back to the reactor for reheating. The results of the analysis indicate that a total of 1882 MW of 1100°C steam can be produced from a 3054 MW power reactor. The steam would be supplied at a rate of 409 kg/s and at a pressure of 1.7 atm. Most of the power (1045 MW) needed to produce the steam comes from a low temperature (121°C) steam generator which can be supplied by the breeding blanket coolant. Approximately 271 MW of steam at 460°C from the first wall is also used. The balance of the power (566 MW) comes from neutron capture within the falling pebble bed. If the steam was recirculated back to the reactor then the low temperature steam generator would be eliminated from the system.

3.1 Thermal Hydraulics Model

A computer code was written to study the behavior of the inner and outer ducts in steady-state and transient operation. With the exception of dimension, the inner and outer ducts, discussed in Sec. 1.1, were analyzed in a similar manner, so no distinction will be made here. Naturally, the results for the inner and outer ducts were different because of size, pellet flow rate, steam flow rate, and neutron heating. Flow rates, weights, and steady-state heat loads for the inner and outer ducts are listed in Tables 3-1 and 3-2, respectively.

Analysis of the pellet regions considered the pellets as forming three parallel columns (Fig. 3-1). A cylindrical coordinate system was used. The control volumes were fixed in space; pellets entered at the top, exchanged heat with their surroundings, gathered heat due to neutron capture and were discharged at the bottom of the control volume. Temperatures varied axially as the pellets move from control volume to control volume within a given vertical column. Temperatures varied radically from column to column. Azimuthal heat transfer within the pellet volume was not considered, however, azimuthal heat transfer to the small radial sides of each duct was included.

Table 3-1. Steady-State Conditions - Inner Duct (Each)

Maximum wall temperature	500°C
Duct weight	809 kg
Total heat load	3.38 MW
Pellet inlet temperature	450°C
Pellet outlet temperature	1157°C
Pellet flow rate	2.47 kg/s
Pellet inventory (assuming a 60% packing fraction)	2451 kg
Pellet transport time	991 s
Pellet transport power expended (assuming a 10-m loss in elevation)	0.25 kW
Pellet heat load via neutron capture	2.94 MW
Heat transport via pellet flow	2.19 MW
Steam heat load	1.19 MW
Wall heat load via sensible heat transfer	0.72 MW
Wall heat load via neutron capture	0.45 MW
Steam inlet temperature and pressure	145°C @ 4 atm
Steam outlet temperature and pressure	461°C @ 3.4 atm
Steam flow rate	1.8 kg/s
Steam inventory	0.325 kg
Steam transport time	180 ms
Steam pumping power expended	none (throttling required)

Five different methods of heat transfer for each of the pellet control volumes were considered:

- (1) Pellet heating due to neutron capture: Volumetric heating rates were calculated based upon the pellet regions being 60% MgO. Although higher densities are theoretically possible, 60% is what is observed.¹¹ The volumetric heating rate is essentially an exponentially decreasing function in the radial direction, which varied from 5.8 to 0.81 W/cm³. Variation in the axial direction was much smaller (~10%) than in radial direction and consequently it was ignored.

Table 3-2. Steady-State Conditions - Outer Duct (Each)

Maximum wall temperature	500°C
Duct weight	1082 kg
Total heat load	16.02 MW
Pellet inlet temperature	450°C
Pellet outlet temperature	1105°C
Pellet flow rate	8.22 kg/s
Pellet inventory (assuming a 60% packing fraction)	6263 kg
Pellet transport time	781 s
Pellet transport power expended (assuming a 10-m loss in elevation)	0.8 kW
Pellet heat load via neutron capture	15.00
Heat transport via pellet flow	13.52
Steam heat load	2.60 MW
Wall heat load via sensible heat transfer	0.69 MW
heat load via neutron capture	0.57 MW
Steam inlet temperature and pressure	145°C @ 4 atm
Steam outlet temperature and pressure	435°C @ 3.2 atm
Steam flow rate	2.05 kg/s
Steam inventory	0.366 kg
Steam transport time	178 ms
Steam pumping power expended	none (throttling required)

(2) Pellet-to-pellet heat transfer: The most important heat transfer means is by thermal radiation. The emissivity of the pellets was assumed to be 0.5. At high temperatures this mechanism dominates. Thermal radiation heat loads to the surrounding Feltmetal-lined walls were considered. Radial heat transfer via thermal conduction among the pellets was also calculated. Reference 12 was used to calculate the effective heat transfer coefficient of the pellets point-to-point constant; the value used was $100 \text{ W/(m}^2\text{-}^\circ\text{K)}$.

(3) Pellet-to-pellet mixing: The model assumed that pellets moved in a vertical direction. In reality, it would be expected that some

Fig. 3-1. Pellet control volume for model.

pellets would migrate from one column to another hence increasing the effective heat transfer coefficient. No data was available for predicting this mixing, so a value equal to the value used for pellet-pellet thermal conduction was used $100 \text{ W}/(\text{m}^2\text{-}^\circ\text{K})$.

- (4) Pellet-to-felt metal: Reference 13 was used to predict the horizontal heat transfer coefficient. Pellets moving down the duct were pressed against the duct walls and made a point contact. The effective heat transfer coefficient used was $200 \text{ W}/(\text{cm}^2\text{-}^\circ\text{K})$.
- (5) Bulk pellet movement: Pellets moved downward into each control volume and consequently convected cold pellets into each removing

some sensible heat. Pellet movement was more complicated near the bottom of the duct as the pellets approached the exit. The pellets over the exit (closest to plasma) moved fastest. Those pellets farther from the plasma had to move horizontally toward the exit. A skewed velocity profile for the bottom three hydraulic diameters was used to account for this movement. An illustration of the assumed flow paths is shown in Fig. 3-2 for the outer duct. A similar distribution was assumed for the inner ducts, however, because the duct was smaller, a control volume was longer than a hydraulic diameter. Consequently the skewed velocity was assumed to exist over two control volumes.

EACH CONTROL VOLUME IS
ONE HYDRALIC DIAMETER LONG

ASSUMED FLOW DISTRIBUTION
IS GIVEN AS A PERCENT OF
TOTAL FLOW

Fig. 3-2. Assumed flow distribution near bottom of outer duct (numbers in percent).

As mentioned earlier, the walls of the duct were hollow and actively cooled with steam. The steam velocity, Reynolds number, pressure, density and heat transfer coefficient were calculated for each control volume as the steam moved upward. The Colburn j-factor was used to calculate the heat transfer coefficient.¹⁴ Thermal communication between the wall and the Feltmetal considered conduction and radiation.

3.2 Numerical Results

Numerical results have revealed much about the proposed design. Cooling of the walls does not appear to be a problem. About 9% of the total heat load of the large outer ducts must be removed by the steam. This fraction increases to 21% for the smaller inner ducts because a larger portion of the total duct volume is occupied by the duct wall. The pellet region was modeled as three parallel columns of pellets. The steady-state temperature varied with radial and axial position, Figs. 3-3 and 3-4. A larger variation in pellet temperature is observed with the outer ducts because there is a large variation in volumetric heating. It was observed that the radial temperature variation was proportional to the axial temperature rise. In order to achieve a mixed pellet temperature of 1100°C, the pellets closest to the plasma source had to be above 1600°K. This resulted in the Feltmetal exceeding the 1000°C limit which is presently considered to be the maximum sustainable temperature in these circumstances. The hottest Feltmetal was found at the bottom inboard side of the outer duct and was at 1075°C. Conversely, the small inner ducts were allowed to operate at a higher temperature because of the small volumetric heating variation. The bulk pellet temperature from the inner ducts is 1158°C. Even with this higher pellet temperature the Feltmetal temperature limit is not exceeded.

Results have shown that startup and shutdown of the reactor would produce very mild transients for the ducts and pellets (see Fig. 3-5). Off-normal events such as a pellet blockage would be quite slow in reaching an excessively high temperature. Normally pellets require 15 min to travel through the duct. If the reactor power remained on, the hottest pellets after 10 min would be ~1800°C, which may result in pellet sintering. If the flow blockage is detected within 2 min and the reactor is shutdown, there would be no structural damage. However, after five minutes the ducts wall are overheating and at a temperature of 560°C.

Fig. 3-3. Pellet temperature within outer duct in steady state.

Fig. 3-4. Pellet temperature within inner duct in steady state.

Fig. 3-5. Mixed pellet reactor outlet temperature for startup and shutdown.

An interruption of steam coolant to the duct walls results in potentially more severe consequences. Transient results are shown in Fig. 3-6 for a 50% and 100% steam blockage without reactor shutdown. With a complete blockage, the duct walls overheat quickly. Within 40 s, the walls have reached 680°C. The reaction time becomes greater when half of the steam still gets through; now two minutes elapse before the walls reach 660°C.

If a detection method were available where a steam blockage would initiate an immediate reactor shutdown then the damage would be lessened. With a 50% steam blockage and a simultaneous reactor shutdown, the walls are slowly cooled and excessive temperatures are never reached, see Fig. 3-7. In fact, if only 25% of the steam is able to get through, the duct wall will be cooled adequately. In the event of a complete steam blockage, a simultaneous reactor shutdown is not enough to save the ducts, see Fig. 3-8. The huge inventory of hot pellets within the duct would melt through the walls. Consequently, a release mechanism would be needed at the bottom of each duct (see Figs. 1-5 and 1-6) to dump the hot pellets in the unlikely event of a complete steam blockage (or loss of steam coolant for some other reason).

Fig. 3-6. Duct wall temperature response after a loss of coolant and failure to scram.
Top: 50% coolant blockage.
Bottom: 100% coolant blockage.

Fig. 3-7. Duct wall temperature after a reduction in coolant and simultaneous scram.

Fig. 3-8. Duct wall temperature after a loss of coolant and simultaneous scram.

3.3 Preheaters

3.3.1 Preheater Design

The present design includes twelve pre-heaters which would warm the pellets from 133°C to 450°C before they move into the inner and outer ducts (see Fig. 1-4). There would be twelve preheaters because there are twelve toroidal field magnets. These components are counter-current direct contact heat exchangers. The pellets enter at the top, are heated and exit at the bottom (see Fig. 3-9). Steam at 460°C and 1.7 atm enters at the bottom through a perforated cone and exits through a perforated region of the dished head at the top. This steam would be in a closed system so that it could be reheated and returned to the pre-heater. The preheater would be insulated to reduce heat loss but there is no need for an active cooling system because the entire vessel would never get above 460°C.

Fig. 3-9. Preheater design.

Each preheater would supply pellets to three inner and three outer ducts for a total flow rate of 56.75 kg/s. There are two reasons for the preheaters: (1) the total megawattage of high temperature heat is increased because now there would be two heat sources - neutron capture within the ducts and sensible heat transfer in the preheater; and (2) the temperature rise within the ducts is reduced and hence the temperature variation at the bottom of the duct is reduced.

The preheaters add a total of 271 extra megawatts to the plant's high temperature capacity. Consequently 271 MW must be available from the reactor sys-

tem. Approximately 409 kg/s of 460°C steam is needed at 1.7 atm for the preheaters. Table 3-3 lists the heat loads and flow rates of the preheaters. This steam could come from the first-wall coolant and/or the duct wall steam coolant. Approximately 600 MW would be available from the first-wall and 87 MW from the duct walls. This steam circuit would require a steam pump to overcome the frictional losses of the steam within the preheater.

The preheater inlet pressure was deliberately chosen to equal the heat-exchanger outlet pressures to minimize steam flow through the connecting falling beds. There would be some neutron activation as the steam flows through the duct. This activation would be proportional to transport time and inversely proportion to the pressure difference between the preheater and heat exchanger. If such a stream were desired it could be accomplished by allowing these pressures to differ. In the present design there is a pressure drop between the heat exchanger bottom and preheater top (0.5 atm). Consequently, there would be some flow through the pellet transport system. However, this is a long path and probably would have a minimal steam flow.

Locating the preheaters directly above the ducts would reduce the overall size of the system, however, this approach would also introduce some structural and maintenance problems. Consequently it was decided to locate the preheaters on a circle larger than the reactor, Fig. 3-10. Note from this figure that although the preheaters and heat exchangers are large, they are not larger than the reactor.

3.3.2 Preheater Analysis

A one-dimensional model was developed to study the preheater design. Pellet and steam temperatures were calculated along the vertical axis of the preheater. Because the pellets required 25 min to travel through the preheaters, an isothermal pellet was assumed. Essentially a linear temperature gradient was calculated for both the pellets and steam. Thermal shock to the pellets is not a problem.

The steam flow resulted in a pressure loss as it moved upward. This pressure drop was calculated using Ref. 12. Several pellet diameters were considered in an effort to reduce the pellet inventory (a 1-cm diameter pellet was finally chosen). There was no definite minimum as a function of pellet diameter. Also investigated was steam pressure and steam flow rate. Again no

Table 3-3. Steady-State Conditions - Preheaters (Each)

Heat load	22.6 MW
Pellet inlet temperature	133°C
Pellet outlet temperature	450°C
Pellet flow rate	56.75 kg/s
Pellet transport time	1476 s
Pellet inventory (assume 60% packing fraction)	50,355 kg
Pellet transport power expended (assuming a 16 m loss in elevation)	8.9 kW
Steam inlet temperature and pressure	460°C @ 1.7 atm
Steam outlet temperature and pressure	142°C @ 1.5 atm
Steam flow rate	34.1 kg/s
Steam transport time	158 ms
Steam inventory	5.41 kg
Steam pumping power expended	4.9 MW

Fig. 3-10. Reactor system plan view.

definite minimum was observed. About the only thing that would reduce the pellet inventory was the pellet-to-steam temperature difference. The ΔT was about 8°C and doubling it would halve the pellet inventory. However, this would result in less megawattage and lower pellet exit temperatures. Because the goal of this study was to maximize the high-temperature megawattage, a large preheater was chosen.

The axial gas pressure drop was always kept lower than that needed to fluidize the bed except at the bottom near the conical inlet. At this elevation the pellets would be held down by the pellets above. The thermal coupling of the pellets and steam was done with a heat transfer coefficient which was calculated axially. A Colburn j -factor was used.¹⁴

3.4 Heat Exchanger Analysis

Twelve heat exchangers are used to heat the process steam from 121°C to 1100°C . Each heat exchanger receives the discharge from three inner and three outer ducts. These devices are similar in design to the preheaters (direct contact counter-current flow), however, they are larger, have a higher heat duty and operating temperature and are actively cooled. Steam enters at the bottom, Fig. 3-11 through a perforated region at two atmospheres and saturation temperature. The steam exits through eight circular holes in the dished head at 1.7 atm. It has been observed that these ceramic pellets do not form a high pile when dumped into a container. Consequently, the relatively shallow dished head will be adequate. Furthermore, the normal pellet operating level with the heat exchanger will be approximately 5 m above the top of the lower conical region. The additional volume is provided so that the heat exchanger can function as a reservoir. The heat exchangers are sized so that all of the pellet inventory can be accommodated within the vessels.

An active cooling system will be necessary to maintain the vessel below 500°C . A cooling capacity of 1.25 MW is estimated for each of the twelve 69.7-MW components. The large volume of these heat exchangers is a direct consequence of the low temperature difference ($\sim 11^{\circ}\text{C}$) between the pellets and steam and the moderate heat transfer coefficient ($\sim 400 \text{ W/m}^2\text{-}^{\circ}\text{C}$). Increasing this ΔT would reduce the volume but would also reduce the final process steam temperature. Table 3-4 lists the steady state heat loads for the heat exchangers.

Fig. 3-11. Heat exchanger design.

Table 3-4. Steady-State Conditions - Heat Exchangers (Each)
Operating Parameters

Heat load	69.7 MW
Pellet inlet temperature	1111°C
Pellet outlet temperature	133°C
Pellet flow rate	56.75 kg/s
Pellet transport time	2488 s
Pellet Inventory (assuming a 60% packing fraction)	141,244 kg
Pellet transport power expended (assuming a 17 m loss in elevation)	9.45 kW
Steam inlet temperature and pressure	121°C @ 2 atm
Steam outlet temperature and pressure	1100°C @ 1.7 atm
Steam flow rate	34.1 kg/s
Steam transport time	430 ms
Steam inventory	14.6 kg
Steam pumping power expended	none (throttling required)

Analysis similar to that done on the preheaters was also done on the heat exchangers. Pellet diameter, pellet flow rate, steam pressure and steam flow rate were varied in an effort to minimize volume. No definite minimum was found.

The long transport time (~41 min) insured that the pellets would cool gradually as they moved through the heat exchanger. The isothermal pellets would obviously not experience any thermal shock. The analysis did not account for heat loss to the actively cooled walls of the heat exchanger. But credit due to afterheat (gamma heating within the heat exchanger due to the induced radioactivity of the ceramic) was ignored and these two heat loads are small and in opposite directions and nearly cancelling.

Reducing the heat exchanger flow area to increase the steam velocity and hence the heat transfer coefficient had mixed results. The goal was to reduce the pellet volume. It soon became apparent that as the flow area was reduced, so was the pellet surface area. Consequently the megawattage diminished. In order to transfer as much heat, a tall narrow heat exchanger would result. No clear advantage was found so the present design remained.

3.5 Alternate Design

During this study, it was assumed that the steam delivered to the synfuel plant was entirely consumed by the synfuel plant. Therefore, of the 3054 MW of thermal power from the reactor, 1882 MW of steam was considered the product. The remaining 1172 MW was available within the reactor system. If the synfuel plant were more intimately connected the reactor system and undecomposed steam rejected by the electrolyzer were returned for reheating, a much more efficient system could result.

The significant changes are: (1) the steam generator would be eliminated; (2) the preheaters would be eliminated; (3) the pellet flow rate would be slightly smaller; (4) the heat exchangers would be smaller; and (5) the pellet inventory would be smaller.

The discharge from the electrolyzer is hot (741°C) oxygen and a hot (741°C) hydrogen-steam mixture. A membrane separator can be used to separate the steam from the hydrogen. Then the hot gases (hydrogen and oxygen) could be used to generate makeup steam to replace the decomposed water eliminating the separate steam generator (1045 MW). This reheated and makeup steam could be combined (474°C) and returned to the heat exchanger.

With this alternate arrangement, the preheaters would be eliminated. The steam entering the heat exchangers is now 474°C and consequently the pellets are never cooled enough to warrant preheating. This reduces the number of components, reduces the pellet inventory and reduces the size of the transport system. Offsetting these advantages are some complications to the pellet transport system. A two-pass system would be needed to supply steam at 1200°C to the synfuel plant. Pellets at 484°C would first travel through the outer ducts and heat to 1109°C. They would then go through the inner ducts to be heated to 1210°C.

A new pellet flow rate would result, now 620 kg/s versus the old 681 kg/s. Because now there would be less megawattage transferred in the heat exchangers, they would be smaller thus further reducing the pellet inventory. The reduced heat exchanger capacity would be 47 MW. However, now the entire heat exchanger vessel would have to be actively cooled. The combined steam flow rate through the heat exchangers would be smaller. The operating pressure would remain at 1.7-atm discharge.

In this arrangement about 566 MW of the fusion power as heat is transferred to the synfuel plant. Another 1250 MW of electric would be needed for the electrolysis process.

Such an arrangement could have several advantages:

- (1) Electric power could be directed to hydrogen production or to the grid as needed.
- (2) Hydrogen production lends itself nicely as a means of energy storage.
- (3) The fusion plant could be operated as a baseline plant similar to a pumped storage system.
- (4) The hydrogen production portion of the plant would operate at an overall efficiency of 58%.
- (5) Oxygen is also a by-product and if it could be used it would constitute an additional advantage.

3.6 Possible Improvements

Analysis of the falling beds revealed that a large variation in pellet temperature occurs within the outer duct. This variation is a result of the uneven heating by neutron capture within the duct. This effect is worse in the outer duct because of its greater thickness and larger variation in neutron fluence. It was observed that the temperature variation was proportional to the total temperature rise along the duct. The large variation gives rise to hot spots on the duct wall.

In the present design there is a 710°C rise in bulk pellet temperature from top to bottom. But there is also a 535°C temperature variation within the pellets at the bottom of the outer duct. A two-pass pellet transport system for the outer ducts would halve the pellet temperature variation at the bottom. There are two important advantages of this change: (1) the maximum temperature of the Feltmetal at the bottom of the outer ducts could be reduced; and (2) the bulk pellet temperature would be increased to 1254°C.

The proposed improvement would first let the pellets flow from the preheaters through half of the outer ducts. The flow from each preheater would be reduced to 46.6 kg/s each but the temperatures would remain the same. The

pellets would exit at 796°C, be lifted to the top of the second half of the outer ducts and continue their heating. At the bottom on the second pass, the bulk pellet temperature would be 1142°C and the peak pellet temperature would be 1254°C. This would eliminate the hot spot on the Feltmetal. The final heating would take place in the inner ducts where the small radial temperature variation is observed. The pellets would exit at 1254°C and the hottest pellet would be 1269°C, thus maintaining the Feltmetal below 1000°C.

There are, however, two disadvantages to this proposed scheme. The transport system would have hot pellets to move. Therefore, a transport system made of ceramic or actively cooled would be necessary. Also the transport system becomes more complex and more power consuming.

Another possible improvement would be a two density ceramic pellet. Let the outer density and hence the thermal conductivity be small in an effort to reduce the sensible heat transfer to the duct walls. If this doesn't do much good, then allow some process steam to flow up through the ducts to the pre-heaters. This steam flow would hopefully eliminate and hot pellet regions by increasing the axial heat flow.

Section 4

STRESS ANALYSIS

4. STRESS ANALYSIS

An extensive structural analysis of the major high temperature blanket components has been conducted. The heat exchanger, the preheater, and both the inner and outer ducts were analyzed. The ASME Boiler and Pressure Vessel Code was used to evaluate the allowable materials stresses and the required structural thicknesses for the components. The stress criteria used in the analysis are reviewed in Ref. 5. The structural material has been assumed to be annealed Type 316 stainless steel in all cases. The analysis did not include the effects of radiation on the mechanical properties.

The general procedure employed consists of determining the various loading forces for each components. The forces are due to internal gas pressure, pellet pressure, the dead weight of the structure, and temperature differences through the structure. These forces are then used to calculate the stresses. The stresses have been divided into primary membrane stresses, local primary stresses, and secondary stresses. Each stress is then compared with the allowable stress intensity, $S_m = 100$ MPa for annealed Type 316 stainless steel at $\sim 500^\circ\text{C}$. The allowable values for the primary membrane, local primary, and secondary stresses are S_m , $1.5 S_m$, and $3 S_m$, respectively. In addition, beam buckling and vibrations were analyzed for both the inner and outer ducts.

In the case of the heat exchanger, the stress criteria can be met with a stainless steel wall thickness of 2.5 cm at the top and sides. The bottom cone of the heat exchanger requires a wall thickness of 2.75 cm. If a corrugated structure is considered, then the side wall thickness can be reduced to 2.1 cm, but the bottom cone thickness remains unchanged. The preheater has a similar configuration to the heat exchanger, and hence, the structural analysis is the same. Because of the preheater's smaller size, however, the required wall thicknesses are reduced. The wall thickness of the top and sides is reduced to 0.8 cm, and the wall thickness of the bottom cone is reduced to 2.0 cm.

The structural analysis for the inner and outer ducts considered a rectangular duct which is 10 m long and constructed from corrugated panels. The details of the corrugated structure is shown in Fig. 4-1. The face plate

Fig. 4-1. Geometry of duct walls used for stress analysis. Symbols are defined in text.

thickness, t_1 and t_2 , is 0.3 cm, and the corrugation plate thickness, t_c , is 0.1 cm. The distance between the face plates, h_c , is 1.6 cm while the pitch of the corrugation, $2p$, is 2.0 cm. Fig. 4-1 also indicates the directions of the bending moments, M_x and M_y , and the tensile forces, N_x and N_y . The structure for the inner duct is shown to easily meet the structural design requirements for both normal and off-normal conditions. During normal operation, the primary stresses are a factor of five below the allowable stress, and the secondary stresses are a factor of two below the allowable stress. During off-normal conditions the primary membrane stress is equal to the allowable stress, but the other stresses are considerably below the allowable stresses. Beam buckling and natural vibrations are shown to be of little concern. In the case of the larger outer duct, two additional ribs have been placed through the structure for additional support (see Fig. 1-6). For this design, all structural requirements are again satisfied.

4.1 Heat Exchanger

The heat exchanger, shown in Fig. 3-11, is a thin-walled compound shell structure consisting of:

- a vertical cylindrical shell with a radius of 2.0 m, length of 8.5 m, and thickness of 2.5 cm;

- the top dished head with a rise of 0.8 m and thickness of 0.25 cm; and

- the bottom conical shell sloped at 45 deg and 2.75 cm thick.

Alternatively the cylinder can be manufactured of corrugated plate with 0.5 cm thick faces and the central distance 2.08 cm. All shells are manufactured of annealed Type 316 stainless steel. The support ring is located at the top of the cylindrical shell in order to avoid compressive stresses and the attendant buckling. Both top and bottom shells are truncated in order to accommodate an inlet and outlet hole of 0.4 m radius. A series of eight smaller holes for steam are uniformly distributed in both the top and bottom shells.

4.1.1 Loading

The loading to which the heat exchanger shells are subjected consists of:

- internal pressure of 2 atm
- pellet pressure
- temperatures of ~500°C.

The pellet pressure is computed according to the ACI standards (Jannsen's method).¹⁵ The maximum lateral pressure turns out to be:

$$p = \frac{\gamma R}{\mu} = 204.2 \text{ kN/m}^2 .$$

where

γ = 20.6 kN/m³, the specific weight of pellets

R = 2.0 cylinder radius

μ = 0.20 coefficient of friction between pellets and wall.

The vertical pressure, q , is then:

$$q = \frac{1 + \sin \delta}{1 - \sin \delta} p = 618.6 \text{ kN/m}^2 ,$$

where δ = 30 deg is the angle of internal friction measured experimentally.

All pressures are multiplied by the larger of the impact factor or the overpressure correction coefficient (see Ref. 15, Art. 4.4.2.2) which turns out to be 2.0 in the present case.

The pressure on the inclined conical surface, q_α , is computed as:¹⁵

$$q_\alpha = p \sin^2 \alpha + q \cos^2 \alpha = 0.5(p + q) = 411.4 \text{ kN/m}^2 .$$

The final pressures (pellets + gas) are thus calculated to be:

	<u>kN/m²</u>
Top dish	$q = 100$
Cylinder	$p = 510$
	$q = 82$
Cone	$q_\alpha = 925$

4.1.2 Analysis

4.1.2.1 Cylindrical Shell

The two types of stresses (according to ASME Boiler Pressure Vessel Code¹⁶) are:

- membrane stresses (in a freely expanding shell);
- discontinuity stresses (arising from the constraint introduced by supports and junctions with other shells); and
- the membrane hoop stresses due to the internal gas and pellet pressure are simply:

$$\sigma_\theta^m = \frac{pR}{t} = \frac{510(2.0)}{t} = \frac{1.02 \times 10^3}{t} \left(\frac{\text{kN/m}}{\text{m}} \right),$$

where t is the thickness of the stainless steel.

The membrane stresses in the axial direction arise from the fact that the entire bottom cone hangs from the cylinder. Estimating the total weight of the structure below the supports, the weight of the pellet inventory (con-

sidering a blocked drain) and the resultant gas pressure on the bottom cone is $W \approx 5500 \text{ kN}$

$$\sigma_x^m = \frac{W}{2\pi R t} = \frac{6500}{2\pi(2.0)t} = \frac{0.517 \times 10^3}{t} \left(\frac{\text{kN/m}}{\text{m}} \right).$$

The determination of the discontinuity stresses is a somewhat more involved process requiring reasonably good knowledge of the elastic shell theory. Nevertheless, since the load is axisymmetric it is possible to derive the formulas for the discontinuity stresses in the present case. The discontinuity stresses are largest if the boundary is built in (i.e. if both radial displacement and rotation are prevented). Assuming that the support ring is much stiffer than the shell this situation indeed exists at the top of the shell.

The maximum discontinuity stresses in a cylindrical shell are, therefore:

- In the case of uniform internal pressure p (note that the pellet pressure is actually slightly less around the top — conservative estimate)

$$\sigma_\theta^d = -(2 - \nu) \frac{pR}{t} = -1.7 \frac{pR}{t}$$

$$\sigma_x^d = - \frac{3(2 - \nu)}{2\sqrt{3}(1 - \nu^2)} \frac{pR}{t} = -1.543 \frac{pR}{t},$$

where $\nu = 0.3$ is the Poisson's ratio.

- In case of a temperature difference $T = T_{\text{shell}} - T_{\text{ring}}$ between the shell and the ring

$$\sigma_\theta^T = -E\alpha T$$

$$\sigma_x^T = - \frac{3}{\sqrt{3}(1 - \nu^2)} E\alpha T = -1.816 E\alpha T,$$

where

$E = 158 \text{ GPa}$, Young's modulus

and

$\alpha = 2.02 \times 10^{-5}/^{\circ}\text{C}$, coefficient of linear thermal expansion.

In summary, using the ASME Boiler and Pressure Vessel Code classification the maximum stress intensities are:

Primary membrane: $P_m = pR/t$

Local primary: $P_L + P_i = 1.700 pR/t$

Secondary: $Q = 1.816 ET$.

The allowed stress intensities and required shell thickness is:

$$P_m = S_m = 1 \times 10^5 \text{ kN/m}^2 ; \quad t_{\text{req}} = \frac{0.51(2.0)}{100} = 0.01 \text{ m}$$

$$P_m + P_L + P_i = 1.5 S_m ; \quad t_{\text{req}} = \frac{2.70(0.51(2.0))}{150} = 0.022 \text{ m}$$

with $t = 2.5 \text{ cm}$ we can now calculate the temperature difference between the shell and the support ring which will result in stresses within the allowable limits

$$T = \frac{Q - (P_m + P_L + P_i)(0.022/0.025)}{1.816(\alpha E)} = 29^{\circ}\text{C} .$$

4.1.2.2 Top Spherical Dished Head

The top head, with a radius of $R_m = 2.91 \text{ m}$, and a central angle of $2\phi_0 = 87 \text{ deg}$ is subjected only to the internal gas overpressure (pellet level is below or at the head-cylinder interface):

- The membrane stresses are simply

$$\sigma_{\theta}^m = \frac{1}{2} \frac{qR_m}{t} = 0.5 \frac{qR_m}{t} .$$

- The discontinuity stresses are determined assuming again the built-in condition at the support ring circumference. In this case, it is again possible to derive the discontinuity stresses in a closed form:

- In case of the uniform internal gas overpressure $p = 100 \text{ kN/m}^2$:

$$\sigma_{\theta}^d = \frac{-3(1-\nu)}{\sqrt{3(1-\nu^2)}} \frac{qR_m}{2t} = -1.271 \frac{qR_m}{2t}$$

$$\sigma_{\theta}^d = (1-\nu) \frac{qR_m}{2t} = -0.70 \frac{qR_m}{2t}.$$

- In case of the temperature difference $T = T_{\text{shell}} - T_{\text{ring}}$:

$$\sigma_{\theta}^T = \frac{3}{\sqrt{3(1-\nu^2)}} E\alpha T = -1.816 E\alpha T$$

$$\sigma_{\theta}^T = -E\alpha T.$$

In summary, the maximum stress intensities are:

Primary membrane: $P_m = 0.5 qR_m/t$

Local primary: $P_L + P_1 = 0.636 qR_m/t$

Secondary: $Q = 1.816 E\alpha T$

The allowed stress intensities and required shell thicknesses are:

$$P_m = S_m = 1 \times 10^5 \text{ kN/m}^2; \quad t_{\text{req}} = \frac{0.10(2.91)}{2(100)} = 0.146 \times 10^{-2} \text{ m}$$

$$P_m + P_L + P_1 = 1.5 S_m; \quad t_{\text{req}} = \frac{(1 + 1.271)(0.10)2.91}{2(150)} = 0.220 \times 10^{-2} \text{ m}$$

with $t = 0.25 \text{ cm}$. We can now calculate the temperature difference (between the shell and the ring) which will keep the stresses within the allowed limits:

$$T = \frac{Q - (P_m + P_L + P_1)(0.22/0.25)}{1.816(\alpha E)} = 29^\circ\text{C}.$$

4.1.2.3 Bottom Conical Shell

The membrane stresses along the largest circumference of the conical shell are:

$$\sigma_{s,max}^m = \frac{1}{2} \left(1 - \frac{\rho^2}{R^2} \right) \frac{qR}{t \cos \alpha} = 0.679 \frac{qR}{t}$$

$$\sigma_{\theta,max}^m = \frac{qR}{t \cos \alpha} = 1.414 \frac{qR}{t},$$

where $\alpha = 45$ deg is the cone angle while $\rho = 0.4$ m and $R = 2.02$ is the radii of the smallest and largest circle.

The discontinuity stresses at the junction with the cylinder are much more difficult to compute. The bending moments and the shear forces at the cylinder-cone intersection can be obtained equating the radial displacements and rotations of the cylinder and the cone at their joint. While these rather cumbersome relations can probably still be derived in the closed form it appears that the thickness of the shell is governed primarily by the primary membrane stresses. In other words, owing to the similar flexibility of the two shells, the discontinuity stresses at the junction are less than 50% of the primary bending stresses. As a result the condition $P_m \leq S_m$ is the governing conditions rather than $P_m + P_L \leq 1.5 S_m$.

In summary from $P_m \leq S_m = 10^5$ kN/m² the required thickness of the required thickness of the bottom crucial shell is

$$t = \frac{1.414 qR}{10^5} = \frac{1.414(925)2.0}{10^5} = 0.026 \text{ m.}$$

Take $t = 2.75$ cm.

As a result of rather large thicknesses there is a temptation to use a corrugated wall with two faces rather than a solid wall. This is possible for the cylindrical case where the combined face thicknesses should support the primary membrane stress, i.e.

$$2t_f = \frac{pR}{S_m} = \frac{1.02 \times 10^3}{10^5} = 0.01 \text{ m}, \quad t_f = 0.5 \text{ cm.}$$

The central distance h between the two face plates should then be determined to support the bending stresses. If the section moduli for the solid and corrugated wall are $(t^2/6)$ and $(t_f h)$ corresponding to the bending moments, then the loads will be safely supported if

$$h = \frac{t^2}{6t_f} = \frac{(2.5)^2}{6(0.5)} = 2.08 \text{ cm}.$$

Unfortunately the corrugated design will not result in any improvement in case of the conical shell since the governing condition is the magnitude of the primary membrane force. Hence, the required thickness is ~2.75 cm (as determined above) regardless of the type of design.

4.2 Preheater

The preheater, shown in Fig. 3-9, is a structure geometrically similar to the heat exchanger. It consists also of the same three thin-walled shell structures:

- a vertical cylindrical shell with a radius of 1.65 m, length of 3.0 m, and thickness of 1.5 cm;
- a top dished head with a rise of 0.6 m and the thickness of 0.8 cm; and
- a bottom conical shell sloped at 450 deg, 2.0 cm thick.

The material of the shell is the annealed Type 316 stainless steel. The support ring is also located at the top of the cylindrical shell. The holes in the top and bottom shell have a 0.33 m radius.

4.2.1 Loading

The loading on the heat exchanger shells consist of:

- internal pressure of 1.5 atm
- pellet pressure
- temperatures of ~500°C.

The pressure on the three parts of the structure are determined exactly as in the case of the heat exchanger. They are found to be:

	<u>kN/m²</u>
Top dish	q = 50
Cylinder	p = 410 q = 82
Cone	q _α = 735

4.2.2 Analysis

The stress analysis follows the steps discussed previously in connection with the heat exchanger. Therefore, the formulas and equations will not be repeated.

4.2.2.1 Cylindrical Wall

The membrane and discontinuity stresses can be determined from the same formulas. Therefore, the required shell thickness,

$$t_{\text{req}} = \frac{0.41(1.65)}{100} = 0.0068 \text{ m}$$

$$t_{\text{req}} = \frac{2.70(0.41)1.65}{150} = 0.0122 \text{ m}$$

with $t = 1.5 \text{ cm}$ the allowable difference in support ring and shell temperature is

$$T = \frac{Q - (P_m + P_L + P_1)(0.0122/0.015)}{1.816(\alpha E)} = 31^\circ\text{C}.$$

4.2.2.2 Top Dished Head

Again the stresses are determined from the same formulas as in the case of the heat exchanger. Hence, the required thickness can be computed from:

$$t_{\text{req}} = \frac{0.05(1.65)}{2(100)} = 0.00041 \text{ m}$$

$$t_{\text{req}} = \frac{(1 + 1.271)0.05(1.65)}{2(150)} = 0.00062 \text{ cm}$$

with $t = 0.8 \text{ cm}$. The allowable difference in support ring and shell temperature is:

$$T = \frac{Q - (P_m + P_L + P_1)(0.00062/0.00080)}{1.816 \alpha E} = 31.7^\circ\text{C}.$$

4.2.2.3 Bottom Conical Shell

The shell thickness is again governed by the membrane stress. Thus,

$$t_{\text{req}} = \frac{1.414 gR}{10^5} = \frac{1.414(735)1.65}{10^5} = 0.017 \text{ m.}$$

Take $t = 2.0 \text{ cm.}$

Since the preheater does not have to be cooled by the steam it is not necessary to consider the corrugated wall structure.

4.3 Inner Duct

4.3.1 Geometry

An inner duct, shown in Fig. 1-5, is structurally a long, thin-walled polygonal shell with a rectangular cross section. The total of 36 geometrically equal ducts are 10 m long while the cross section is approximately a rectangle 76 cm wide and 20 cm deep. The sides of the shell are manufactured from the annealed Type 316 stainless steel in the form of corrugated sheets. The face plates of the sheets are 0.3 cm thick while the corrugation is only 0.1 cm thick. The distance between the centers of two face plates is 1.6 cm while the pitch of the corrugation is 2.0 cm.

The stiffness parameters of the corrugated sheets are computed according to Ref. 17.

The extensional and bending stiffness along the corrugation (vertically) are:

$$A_x = (1.282)2t_f$$

$$I_x = 1.12 t_f h / 12 ,$$

where $t_f = 0.3 \text{ cm}$ (face plate thickness) and $h = 1.6 \text{ cm}$ (central distance between face plates).

The extensional and bending stiffnesses across the corrugation (horizontally) are:

$$A_y = 0.943(2 t_f)$$

$$I_y = 1.01(1/2 t_f h^2) .$$

The shear and tortional stiffnesses are not necessary at this stage of the analysis.

4.3.2 Loading

The loading on the duct consists of:

- pellet pressure
- difference in gas pressures
- temperature difference
- dead weight.

The maximum gas pressure in the duct is 2.0 atm. The steam pressure inside the corrugated walls is 5.0 atm. The pressure outside the ducts is atmospheric.

The pellet pressure is computed according to the already listed Jannsen's formula according to which the horizontal pressure in the walls of the duct is:¹⁵

$$p = \frac{\gamma R}{\mu} \left[1 - \exp - \left(\frac{\mu' k y}{R} \right) \right] = \frac{\gamma R}{\mu} , \quad \text{for } y > \frac{R}{\mu' k} ,$$

where y is the depth of the pellets measured from the top. The hydraulic radius R is in case of the rectangular container given by:

$$R = \frac{1}{2} \frac{ab}{a + b} ,$$

where $a = 0.76$ m and $b = 0.20$ m are the sides of the rectangle. For $\mu = 0.30$, $\gamma = 20.6$ kN/m³, and $R = 0.08$ m the horizontal pressure is:

$$p_{\max} = 5.5 \text{ kN/m}^2 .$$

The vertical pressure is:

$$q = \frac{1}{k} p = \frac{1 + \sin \delta}{1 - \sin \delta} p = 16.5 \text{ kN/m}^2$$

for $\delta = 30$ deg. The larger of the overpressure correction factor and the impact factor is 2.0. In addition, according to Article 4.4.2.4 of Ref. 15,

it is necessary to increase the pressure because of the eccentric discharge. Although the commentary to Ref. 15 suggests the increase of ~25% in this analysis a conservative 50% increase was used. Thus,

Maximum vertical pressure: $q = 16.5(2.0)1.5 = 49.5 \text{ kN/m}^2$

Maximum horizontal pressure: $p = 5.5(2.0)1.5 = 16.5 \text{ kN/m}^2$

Maximum vertical frictional force: $v = (\gamma y - 0.8 q)R(2.0)1.5 = 36.4 \text{ kN/m}^2$.

The maximum temperature difference between two walls was taken to be 20°C; while the maximum temperature drop across the wall was taken as 50°C.

In summary, analysis considers two service loading levels:

- Level A service loading -- pressure of the pellets conveyed through the duct and the computed temperature differences.
- Level C service loading -- infrequently occurring incidents (internal pressure difference of 2.00 atm).

4.3.3 Stress Analysis

The duct is structurally a polygonal shell. However, since the sides of the duct are relatively stiff it is considered that sufficient accuracy can be obtained if:

- the hoop stresses are determined from the bending of the rectangular frame (cross section), while
- the axial stresses are computed from the conventional beam analysis.

4.3.3.1 Hoop Stresses

- Bending of a rectangular frame subjected to the uniform internal pressure p_h . The maximum bending moment, M_p , (occurring at the vertex) and the maximum normal (tensile) forces are:

$$M_p = \left(1 - \frac{2}{3} \frac{1 + \beta^3}{1 + \beta} \right) \frac{p_h a^3}{8}$$

$$N_p = \frac{1}{2} p_h a ,$$

where $\beta = b/a = (0.20/0.76) = 0.2633$ is the length aspect.

- Temperature difference between the walls. If the temperature of the wall towards the plasma is T_1 and the temperature of the other (parallel) wall is T_2 the maximum bending moment is:

$$M_{T_1} = \alpha(T_1 - T_2) \frac{EI}{b} \frac{1 + \beta}{(1 + 2/3 \beta)^2 - 1/9 \beta^2}$$

$$= 2.919 \times (T_1 - T_2) \frac{EI}{b}.$$

- Temperature drop across the wall. If the temperature drop across the walls is ΔT , then the maximum bending moment is:

$$M_{\Delta T} = \frac{(\alpha \Delta T) EI}{h} \frac{(1 + \beta)(1 + 1/3 \beta)}{(1 + 2/3 \beta)^2 - 1/9 \beta^2} = \frac{(\alpha \Delta T) EI}{h}.$$

- Local bending of the face plates. Considering the face plate to be a continuous beam (of unit width) with spans equaling the pitch of the corrugation, $l = 2.0$ cm, the maximum bending moment and the shear force are:

$$M_{L, \max} = \frac{p_t l^2}{12} \text{ (at supports)}$$

$$V_{L, \max} = \frac{1}{2} p_f l,$$

where p_f is the pressure difference.

- For the steam pressure inside the corrugated wall the corrugation itself keeps the face plates together. Considering the cross section of a corrugated plate as a truss, the force in the truss diagonal (sloped at $\theta = 60$ deg to the face plate) is:

$$N_{\max} = \frac{P}{2 \sin \theta} = \frac{P}{1.732} \text{ (tensile),}$$

where $P = p_f l$ (N/m) is the pressure difference concentrated at the truss node.

The hoop stresses can now be computed and clarified again according to the ASME Boiler and Pressure Vessel rules.¹⁶ In the present case:

- Primary membrane stress intensity P_m due to the uniform internal pressure p_h

$$\sigma_{y,\max}^{(M)} = \frac{M_P}{I_y} \frac{h + t_f}{2} = 0.806 \frac{p_h a^2}{12} \frac{2}{1.01 t_f h^2} \left(\frac{h + t_f}{2} \right) = 950 p_h$$

$$\sigma_y^{(N)} = \frac{N_P}{A_y} = \frac{p_h a}{2} \frac{1}{0.973(2 t_f)} = 67 p_h,$$

both in units of the pressure p_h . Thus, noting that even $\sigma_y^{(M)}$ is a membrane stress at each face plate

$$P_m = (950 + 67) p_h = 1017 p_h.$$

- Primary bending stress intensity P_b , due to the steam pressure inside the wall:

$$\sigma_{\max} = \frac{M_{L,\max}}{S_y} = \pm \frac{p_f l^2}{12} \frac{6}{t_f^2} = \pm 22.2 p_f = P_b$$

in units of the pressure difference p_f across the face plate. Also, the shear stress is:

$$\tau_{\max} = 1.5 V_{L,\max}/t_f = 5.0 p_f$$

and normal stress in the corrugation is:

$$\sigma_{\max,c} = \frac{l p_f}{1.732 t_c} = 11.54 p_f = P_m.$$

- Secondary stress intensity Q due to the temperature differences

$$\begin{aligned} \sigma^{(T_1-T_2)} &= 0.919 \alpha (T_1 - T_2) \frac{E I_y}{b} \frac{1}{1.01 I_y} \frac{h + t_f}{2} \\ &= 0.039 \alpha (T_1 - T_2) E, \end{aligned}$$

$$\sigma_y^{\Delta T} = \frac{\alpha \Delta T}{h} \times \frac{1}{1.01 I_y} \left(\frac{h + t_f}{2} \right) = 0.526 \alpha \Delta T E ,$$

both in units of E. Thus, the secondary stress intensity is

$$Q = [0.039(T_1 - T_2) + 0.526 \Delta T] \alpha E .$$

4.3.3.2 Beam Stresses

The duct is assumed to be simply supported at the top and the bottom allowing for the unrestrained axial thermal expansion. The moment of inertia of the corrugated box section is:¹⁸

$$\begin{aligned} I_b &= 2a(1.282)2t_f \left(\frac{b}{2} \right)^2 + 2 \frac{(1.282)2t_f}{12} b^3 \\ &= at_f b^2 (1.282) \left(1 + \frac{1}{3} \frac{b}{a} \right) = 12717 \text{ cm}^4 , \end{aligned}$$

where $t_f = 0.3 \text{ cm}$, $a = 76 \text{ cm}$, and $b = 20 \text{ cm}$. Thus, the section modulus is

$$S_b = I_b \frac{2}{b} = 2at_f b (1.282) \left(1 + \frac{1}{3} \frac{b}{a} \right) = 1271 \text{ cm}^3 .$$

The cross sectional area is:

$$A_b = 2(a + b)1.282(2t_f) = 148 \text{ cm}^2 .$$

The loading on the duct as a beam consists of the difference between the pressures in the "hot" (closer to plasma) and the "cold" wall. Since the space encircled by the ducts is not sealed off there cannot be any differential gas pressure. The only differential pressure which can occur is due to the off-center discharge of the pellets. Assume conservatively that the difference between the pellet pressures on the front and back wall is:

$$P_b = 16.5(0.76) = 12.54 \text{ kN/m}^2$$

(i.e. that the pressure on the back wall practically vanishes.

Assume conservatively that this pressure difference occurs in full amount at the bottom and then linearly decreases over a length of one-fourth of the span (approximately 30 hydraulic radii). The maximum moment occurs at

$$y = (\ell_1 + \ell_2 \sqrt{\ell_2/3L}) = 0.822 L ,$$

where $\ell_1 = 3/4 L$ and $\ell_2 = 1/4 L$. Then the maximum bending moment is:

$$M_{b,max} = \left[\frac{\ell_2^2 y}{6L} - \frac{1}{6\ell_2} (y - \ell_1)^3 \right] p = 0.00832 pL^2 = 10.4 \text{ kn-m} .$$

Hence, the bending stresses in the beam (which are primary membrane stresses for the face) are:

$$\sigma_{x,max}^{(b)} = \frac{M_{b,max}}{S_b} = \frac{10.4}{1271} \times 10^6 = 0.08 \times 10^5 \text{ kN/m}^2 .$$

The normal force associated with the dead weight of 809 kgt and the pellet inventory of 2451 kg (assuming the friction coefficient of 0.3) is:

$$N_{b,max} = [809 + 0.3(2451)]9.8066 = 15144 \text{ N} .$$

Thus, the corresponding normal stress is:

$$\sigma_{x,max}^{(n)} = \frac{15.144}{148} 10^4 = 1.02 \times 10^3 \text{ kN/m}^2 .$$

Consequently, the primary membrane stress intensity on the duct face plate in the axial direction is

$$P_m^x = 0.09 \times 10^5 \text{ kN/m}^2$$

4.3.3.3 Summary of Stresses

Level A Service Loading

- Primary membrane stress intensity due to the pellet pressure only ($p_h = 16.5 \text{ kN/m}^2$) :

$$P_m = 0.09 \times 10^5 + 16.5(1017) = 0.26 \times 10^5 \text{ kN/m}^2 .$$

- Primary bending stress intensity (due to the steam pressure inside the wall — steam is at 5 atm maximum):

$$P_b = 22.2(4)1.015 \times 10^2 = 0.09 \times 10^5 \text{ kN/m}^2 .$$

- Secondary stress intensity due to the temperature difference of $T_1 - T_2 = 720^\circ\text{C} - 709^\circ\text{C}$ and the temperature drop of 50°C across the wall:

$$\begin{aligned} Q &= [0.039(20) + 0.526(50)](2.02 \times 10^{-5})1.58 \times 10^8 \\ &= 0.86 \times 10^5 \text{ kN/m}^2 . \end{aligned}$$

Thus,

$$P_m = 0.26 \times 10^5 \text{ kN/m}^2 < S_m = 1.06 \times 10^5 \text{ kN/m}^2$$

$$P_m + P_b = 0.35 \times 10^5 \text{ kN/m}^2 < 1.5 S_m$$

$$P_m + P_b + Q = 1.21 \times 10^5 \text{ kN/m}^2 < 3.0 S_m .$$

Level C Service Loading

- Primary membrane stress intensity due to the pellet pressure and the internal pressure of 2.0 atm ($p_h = 16.5 + 101.3 = 117.8 \text{ kN/m}^2$):

$$P_m = 0.09 \times 10^{-5} + 117.8(1017) = 1.29 \times 10^5 \text{ kN/m}^2 .$$

- The primary bending and secondary stress intensity remain unchanged:

$$P_b = 0.09 \times 10^5 \text{ kN/m}^2 , \quad Q = 0.86 \times 10^5 \text{ kN/m}^2 .$$

Thus,

$$P_m = 1.29 \times 10^5 \text{ kN/m}^2 = 1.25 S_m = 1.27 \times 10^5 \text{ kN/m}^2$$

$$P_m + P_b = 1.38 \times 10^5 \text{ kN/m}^2 < 3 S_m = 3.18 \times 10^5 \text{ kN/m}^2$$

$$P_m + P_b + Q = 2.25 \times 10^5 \text{ kN/m}^2 < 3 S_m = 3.18 \times 10^5 \text{ kN/m}^2 .$$

4.3.4 Stability

As a vertical thin-walled column, a duct like this is, in general, susceptible to buckling. Several buckling modes will be briefly examined.

4.3.4.1 Beam Buckling

The critical buckling force for a simply supported column is:

$$N_{cr} = \frac{\pi^2 EI_b}{L^2} = \frac{\pi^2 (1.58 \times 10^{11}) 1.1695 \times 10^{-4}}{10^2} = 1.82 \times 10^6 \text{ N}.$$

The maximum force $N_{b,max}$ is only $1.51 \times 10^4 \text{ N}$ (Sec. 4.3.3.2). Thus, even if the entire weight was concentrated at the top, $N_{cr} \gg N_{b,max}$.

4.3.4.2 Buckling of the Corrugated Plate Using the Formulas from Ref. 18

$$D = \frac{Et^3}{2(1 - \nu^2)} (1 + h/t)^2 = 22.04 Et^3$$

$$J = \frac{G hb^2}{D \pi^2} = \frac{1}{2(1 + 0.3)} \frac{1.6(76.0)^2}{22.04 \pi^2 (0.3)^2} = 605.2.$$

From the graph in Ref. 18 $K_x = 4.0$ since

$$\frac{a}{b} = \frac{1000}{76} > 3.0.$$

Finally,

$$N_{x,cr} = \frac{\pi^2 DK_x}{b^2} = 1.6 \times 10^6 \text{ N/m}.$$

If the entire force $N_b = 1.51 \times 10^4 \text{ N}$ is uniformly divided along the circumference

$$N_x = \frac{1.51 \times 10^4}{2(0.2 + 0.76)} = 0.79 \times 10^4 \frac{\text{N}}{\text{m}} \ll N_{x,cr}.$$

4.3.4.3 Buckling of the Face Plate

Considering the spanning the corrugation pitch as a freely supported

$$\frac{N_{cr}}{A} = \frac{\pi^2 EI}{l^2 A} = \left(\frac{\pi t_t}{l} \right)^2 \frac{E}{12} = 3.25 \times 10^5 \text{ kN/m}^2 \geq S_m ,$$

i.e. the face plate will yield before it buckles.

In conclusion, in all cases there is a large safety factor against buckling (both global and local).

4.3.5 Natural (Free) Vibrations

4.3.5.1 Beam Vibration

The lowest natural frequency of transverse (general) vibrations is:

$$\omega_t = \frac{\pi^2}{L^2} \sqrt{\frac{EI}{m}} ,$$

where $\omega = 77 \times 10^3 A_b = 77.0(148.0)10^{-1} = 1140 \text{ N/m}$, such that $m = w/g = 1140/9.81 = 116 \text{ N-s}^2/\text{m}^2$. Hence,

$$\omega_t = \frac{\pi^2}{10^2} \sqrt{\frac{1.58 \times 10^{11}(1.2717) \times 10^{-4}}{116}} = 41.1 \text{ rad/s} .$$

The period of free vibrations is thus:

$$T_t = \frac{2\pi}{\omega_t} = 0.15 \text{ s} .$$

4.3.5.2 Column Vibration

The lowest natural frequency of axial vibrations is given by:

$$\omega_a = \frac{\pi}{L} \sqrt{\frac{Eg}{\gamma}} = \frac{\pi}{10.0} \sqrt{\frac{1.58 \times 10^{11}(9.81)}{77.0 \times 10^3}} = 0.141 \times 10^4 \text{ rad/s} ,$$

where $\gamma = 77.0 \text{ kN/m}^3$. Thus, the natural period is:

$$T_a = \frac{2\pi}{\omega_a} = 4.45 \times 10^{-3} \text{ s}.$$

4.3.5.3 Cross-Sectional Vibrations

Considering the long side of the cross section as simply supported the natural frequency is:

$$\omega_{cs} = \frac{\pi^2}{a^2} \sqrt{\frac{EI_x}{m}},$$

where

$$W = 77.0 \times 10^3 (2 \times 0.003 \times 1.282) = 592 \text{ N/m}$$

$$m = \frac{592}{9.81} = 60.3 \frac{\text{N-s}^2}{\text{m}^2},$$

hence (see Sec. 4.3.1):

$$I_x = 1.12 \frac{t_f h^2}{12} = 0.717 \times 10^{-7} \text{ m}^4/\text{m}.$$

Hence,

$$\omega_{es} = \left(\frac{\pi}{0.76} \right)^2 \sqrt{\frac{1.58 \times 10^{11} (0.717 \times 10^{-7})}{60.3}} = 234 \text{ rad/s}$$

and

$$T_{cs} = \frac{2\pi}{\omega_{cs}} = 2.69 \times 10^{-2} \text{ s}.$$

4.3.5.4 Face Plate Spanning the Corrugation Pitch

The free vibration frequency is:

$$\omega_f = \left(\frac{\pi}{l} \right)^2 \sqrt{\frac{EI_f}{m_f}},$$

where

$$m_f = \frac{77.0 \times 10^3 (0.003)}{9.81} = 23.55 \text{ N-s}^2/\text{m}^2$$

$$I_f = \left(\frac{\pi}{0.02} \right)^2 \sqrt{\frac{1.58 \times 10^{11} (2.25 \times 10^{-4})}{23.55}} = 9.59 \times 10^4 \text{ rad/s}$$

and

$$T_f = \frac{2\pi}{\omega_f} = 0.66 \times 10^{-4} \text{ s}.$$

In summary, only the beam bending vibration natural frequency is low enough to cause some concern. All other frequencies are too high to result in significant dynamic amplification factors.

4.4 Outer Duct

4.4.1 Geometry

Owing to a larger circumference along which the 36 outer ducts are located the dimensions of the cross section are increased to $1.705 \times 0.40 \text{ m}$. Hence, in order to use the same corrugated walls as in the case of the inner ducts the cross-sectional box must be adequately strengthened.

Since the preliminary analyses indicated that a large $1.705 \times 0.40 \text{ m}$ box manufactured of the same corrugated walls as the inner ducts will not be structurally adequate three other possibilities were considered:

- doubling the number of ducts, ~72 (i.e. using a duct with a rectangular cross section measuring $0.85 \times 0.40 \text{ cm}$);
- using a rib in the module (i.e. a two-cell cross section); and
- using two ribs dividing the cross section into three equal cells.

The tradeoff in the design is apparent from Table 4-1 using the large $1.704 \times 0.40 \text{ m}$ duct as the datum. Hence, the addition of two ribs converting the duct into a triple cell design adds 19% of steel, reduces the cross section available for pellets by 14%, but reduces the stress in the cross section to only 15% of the original. As a result the tripple cell design was judged as being the most promising one for the present circumstances.

Table 4-1. Design Comparison for the Outer Duct

	Single Cell		Double Cell	Triple Cell
	36 Ducts	72 Ducts	36 Ducts	36 Ducts
$\sigma_y^{\max}/(\sigma_y^{\max})_{36}$	1	0.236	0.317	0.150
A/A_{36}	1	1.190	1.095	1.190
$A_{\text{net}}/(A_{\text{net}})_{36}$	1	0.976	0.988	0.976

The extensional and bending stiffnesses of the corrugated wall are the same as as in the case of the inner duct (see Sec. 4.3.1).

4.4.2 Loading

The loading is essentially the same as in the case of the inner duct with several small differences. As a result of the smaller hydraulic radius,

$$R = \frac{1}{4} (0.47) \approx 0.12 \text{ m} .$$

The horizontal and vertical pellet pressures computed according to the Jannsen formula (see Sec. 4.3.2) are:

$$p_{\min} = 7.5 \text{ kN/m}^2$$

$$q_{\max} = 15.0 \text{ kN/m}^2$$

Hence, after multiplying with all overload coefficients:

$$q = 75 \text{ kN/m}^2$$

$$p = 22.5 \text{ kN/m}^2 .$$

The maximum difference between the temperatures of the "hot" and "cold" wall is $748^\circ\text{C} - 713^\circ\text{C} = 35^\circ\text{C}$.

4.4.3 Stress Analysis

4.4.3.1 Hoop Stresses

- Bending of the three-cell frame subjected to the uniform internal pressure p_h . The maximum bending moment (occurring in the face plate just inside of the stiffening rib) is:

$$M_p = \frac{1 + 7\beta + 6\beta^2 - 2\beta^3 - \beta^4}{5\beta^2 + 5\beta + 1} \frac{p_h a_1^2}{12} = 1.138 \frac{p_h a_1^2}{12},$$

where $a_1 = 0.57$ m and $\beta = 0.40/0.57 = 0.702$. The maximum tensile force (in the middle rib) is:

$$N_p = 2 \frac{p_h a_1}{2} + \frac{1.087 - 0.719}{a_1} \frac{p_h a_1^2}{12} = 1.031 p_h a_1.$$

- Temperature difference between the walls and temperature drop across the thickness. Using the same approach as in Sec. 4.3.3.1 with new β it follows that:

$$M_{T_1} = 0.927 \alpha (T_1 - T_2) \frac{EI_y}{b}$$

and

$$M_{\Delta T} = \frac{(\alpha \Delta T) EI_y}{h}.$$

- Local bending of the face plates. Since the same corrugated wall is used the maximum bending moment and shear force are again:

$$M_{L,max} = \frac{p_f \ell^2}{12} \quad (\text{at supports})$$

$$V_{L,max} = \frac{1}{2} p_f \ell.$$

The same is true for the normal force in corrugation which is:

$$N_{\max} = \frac{p_f \ell}{1.732}.$$

The hoop stresses are now computed and classified in the same way as in Sec. 4.3.3.1, i.e.

- Primary membrane stress intensity, P_m , due to the uniform internal pressure p_h :

$$\sigma_{y,\max}^{(m)} = \frac{M_P}{I_y} \frac{h + t_f}{2} = 1.138 \frac{p_h a_1^2}{12} \frac{2}{1.01 t_f h^2} \frac{h + t_f}{2} = 754.7 p_h$$

and

$$\sigma_y^{(N)} = \frac{N}{A} = 1.031 \frac{p_h a_1}{A} = 51.9 p_h.$$

Therefore,

$$P_m = (754.7 + 51.9) = 806.6 p_h$$

in units of the pressure p_h .

- Primary bending stress intensity P_b , due to the steam pressure, is as in the case of the inner duct

$$P_b = 22.2 p_f.$$

In corrugation (the primary membrane stress intensity) is:

$$P_m = 11.54 p_f.$$

- Secondary stress intensity Q due to the temperature difference is:

$$Q^{(T_1-T_2)} = 0.020 \alpha (T_1 - T_2) E$$

$$Q^{\Delta T} = 0.526 (\alpha \Delta T) E$$

in units of the elastic modulus E .

4.4.3.2 Beam Stresses

The duct is assumed to be simply supported at the top and the bottom allowing for the unrestrained thermal expansion. The moment of inertia of the three-cell corrugated box is:

$$\begin{aligned} I_b &= 2a(1.282)2t_f(b/2)^2 + 4 \frac{(1.282)2t_f}{12} b^3 \\ &= 1.182 a t_f b^2 \left(1 + \frac{2}{3} \frac{b}{a}\right) = 121328 \text{ cm}^4, \end{aligned}$$

where $t_f = 0.3 \text{ cm}$, $a = 170.5 \text{ cm}$, and $b = 40 \text{ cm}$. The section modulus is:

$$S_b = I_b(2/b) = 121328(2/40) = 6066 \text{ cm}^3.$$

The cross-sectional area is:

$$A_b = (2a + 4b)1.282(2t_f) = 385 \text{ cm}^2.$$

Using again the same conservative assumptions regarding the imbalance of the pellet pressure associated with the off-center discharge it follows that:

$$P_b = (1.705)22.5 = 38.36 \text{ kN/m},$$

while

$$M_{b,\max} = 0.00832 pL^2 = 31.9 \text{ kN-m}.$$

Hence, the bending stresses in the beam:

$$\sigma_{x,\max}^{(b)} = \frac{M_{b,\max}}{S_b} = \frac{31.9}{6066} \times 10^6 = 0.05 \times 10^5 \text{ kN/m}^2.$$

Estimating the normal stress due to the normal force to be the same as in the case of the inner duct the primary membrane stress intensity on the duct face plate in the axial direction is:

$$P_m = 0.06 \times 10^5 \text{ kN/m}^2.$$

4.4.3.3 Summary of Stresses

Level A Service Loading

- Primary membrane stress intensity due to the pellet pressure only ($p_h = 22.5 \text{ kN/m}^2$):

$$P_m = 0.06 \times 10^5 + 22.5(754.7) = 0.23 \times 10^5 \text{ kN/m}^2 .$$

- Primary bending stress intensity (due to the steam at 5 atm in the corrugated walls):

$$P_b = 22.2(4)1.015 \times 10^2 = 0.09 \times 10^5 \text{ kN/m}^2 .$$

- Secondary stress intensity due to the temperature difference of 35°C between the two walls and the temperature drop of 50°C across the walls:

$$\begin{aligned} Q &= [0.020(35) + 0.526(50)](2.02 \times 10^{-5})1.58 \times 10^8 \\ &= 0.86 \times 10^5 \text{ kN/m}^2 . \end{aligned}$$

Thus,

$$P_m = 0.23 \times 10^5 \text{ kN/m}^2 < S_m = 1.06 \times 10^5 \text{ kN/m}^2$$

$$P_m + P_b = 0.32 \times 10^5 \text{ kN/m}^2 < 1.5 S_m$$

$$P_m + P_b + Q = 1.18 \times 10^5 \text{ kN/m}^2 < 3 S_m .$$

Level C Service Loading

- Primary membrane stress intensity due to the pellet pressure and the internal pressure of 20 atm ($p_h = 22.5 + 101.3 = 123.8 \text{ kN/m}^2$):

$$P_m = 0.006 \times 10^5 + 123.8(754.7) = 0.99 \times 10^5 \text{ kN/m}^2 .$$

- The primary bending and the secondary stress intensity remain unchanged:

$$P_b = 0.09 \times 10^5 \text{ kN/m}^2 , \quad Q = 0.86 \times 10^5 \text{ kN/m}^2 .$$

Thus,

$$P_m = 0.99 \times 10^5 \text{ kN/m}^2 < 1.25 S_m = 1.27 \text{ kN/m}^2$$

$$P_m + P_b = 1.08 \times 10^5 \text{ kN/m}^2 < 1.85 S_m$$

$$P_m + P_b + Q = 1.94 \times 10^5 \text{ kN/m}^2 < 3 S_m .$$

4.4.4 Stability

The outer duct is as a result of two additional ribs stiffer than the inner duct. In view of large safety margins computed for the inner duct the stability analysis for the outer duct is unnecessary.

Appendix A

PEBBLE FLOW EXPERIMENT

A.1. Introduction

One of the major problems in the design of a high temperature flow pebble blanket is the nonuniformity of the neutron power density. The volumetric heating ratio varies from 5.8 W/cm^3 near the front of the blanket to 0.81 W/cm^3 near the back of the blanket. If the flow rate is constant through the blanket, then the pebbles near the front will reach much higher temperatures than the pebbles near the rear of the blanket. From a design point of view this temperature gradient is unacceptable, since the average temperature of the pellets exiting the blanket will be low even though the maximum temperature limits are reached near the front of the blanket. In order to equalize the temperatures it is desirable to vary the flow rate such that the pellets receiving the lowest volumetric heating have the lowest velocity. Under ideal conditions, all the pellets at the bottom of the duct would reach the highest allowable temperature independent of their radial position. The method chosen to vary the flow rate in the present design is to position the exit for the pellets near the front of the duct to enable the pellets at the front to flow at the fastest rate. Unfortunately, there is little experimental information with which to predict pellet flow and mixing rates. An experiment was therefore undertaken to examine the flow characteristics through a rectangular duct with an off-centered rectangular exit.

The primary purposes of the experiment were to establish a technique for quantitatively examining the flow characteristics of a falling pebble bed and to perform a limited number of tests to determine the viability of the duct and exit design used in the production of high temperature steam. Since resources were limited, an extensive investigation was not conducted.

A.2 Experimental

The duct used in the experiment is shown in Fig. A-1. The size of the duct is 0.23 m wide $\times$ 0.15 m deep and 1.5 m long. The duct was filled with Al_2O_3 pellets $\sim 1 \text{ cm}$ diameter, and the pellets were allowed to fall out of the duct through an off-centered rectangular exit. The exit size was approximately

Fig. A-1. Experimental layout for pebble flow measurements.

one-fourth of the area of the duct and the size was held constant during the test runs. The slope from the back of the duct to the exit was varied with angles of 33, 45, and 63 deg being used for the tests. The desired information was the pellet velocity at the front and rear of the duct and the degree of mixing that occurs as the pellets fall through the duct.

The velocity and mixing of the pellets was determined using a radioactive counting technique. The detection equipment consisted of a 300-mC ^{137}Cs gamma-ray source (550 keV) and a gamma-ray detector which was connected to a 512-channel analyzer. As the pellets fell between the ^{137}Cs source and detector they produced an attenuated counting rate at the detector. The gamma-ray counting rate was recorded with the multichannel analyzer, and the numerical

results were then placed in computer files for later analysis. Each channel corresponds to a time increment of 50 ms.

The flow characteristics of pellets from a particular region of the duct were determined by the use of tracer pellets. The tracer pellets were composed of steel and were the same size as the ceramic pellets. Because of the higher density of the steel pellets, they produced a higher attenuation factor for the gamma rays. By locating the gamma beam at the exit, the passage of the steel balls could be detected by a reduced counting rate at the gamma-ray detector. In order to translate the counting rate data into velocity and mixing information, several assumptions were made concerning the nature of the attenuation and flow characteristics of the pebbles:

- (1) The steel pellets are assumed to have the same basic flow characteristics as the ceramic pellets. A concern in the experiment is that the heavier steel pellets would fall at a faster rate than the ceramic pellets. Visual observations of the flowing pellet bed indicated that the pellet mixing was of a random nature. Therefore, it is believed that contact friction forces and pressure within the duct, and not the pellet density, determine the motion characteristics.
- (2) It is assumed that the gamma-ray attenuation follow an exponential form.
- (3) It is assumed that the attenuation factor of the ceramic and steel is directly related to the actual mass within the gamma-ray beam. This assumption allows the use of superposition to determine the fraction of tracer pellets within the falling stream.

Using the above assumptions the general form of the counting rate at the detector is given by:

$$A = A_0 \exp(-\lambda m) \quad (A.1)$$

where A is the count rate at the detector, A_0 is the count rate at the ^{137}Cs source, λ is an attenuation coefficient which is a combination of the attenuation coefficients for ceramic pellets, λ_c , and steel pellets, λ_s , and m is the packing factor of the pellets.

The experiment is divided into two parts: calibration tests to establish the values of λ and μ for different combinations of ceramic and steel pellets; and tests to determine the mixing and flow characteristics of tracer pellets at various locations and for different exit slopes in the duct.

A.3 Results

A summary of all the test runs is presented in Table A-1.

Table A-1. Summary of Tests

Date	Test No.	Description	Average (rms)
8/26/81	1	Background counts	59.4(7.79)
	2	Static steel	141.5(11.7)
	3	Static 50% steel-50% ceramic	225(14)
	4	Static ceramic	555(23)
	5	Unattenuated source with bracket	7505(49)
	6	Unattenuated source, no bracket	7643(52)
	7	Falling ceramic	3867(218)
	8	Falling steel	1472(159)
	9	Falling 50% steel-50% ceramic	2441(289)
	10	Steel front from 4.75 to 8.5 in.	
	11	Steel back from 4.75 to 8.5 in.	
	12	Background	
	13	Unattenuated source	7641(48)
8/27/81	1	All steel	2550
	2	Bottom 5.25 in. is steel	
	3	Steel front from 4.75 to 8.25 in.	
	4	Steel back from 5 to 9 in.	
8/28/81	1	Bottom 7.5 in. is steel	
	2	Steel front from 7.65 to 12 in.	
	3	Steel back from 7.5 to 11.75 in.	
	4	Background	
	5	Falling ceramic	60(7.4)
	6	Unattenuated source	3714(212) 7570(53)
8/26/81	Exit geometry: 45-deg angle, 1-5/8 in. wide opening		
8/27/81	Exit geometry: 33-deg angle, 1-1/2 in. wide opening		
8/28/81	Exit geometry: 63-deg angle, 1-1/2 in. wide opening		

A.3.1 Calibration

The calibration tests consisted of first determining λ_s and λ_c , then determining the volumetric density of pellets falling out of the duct for the various exit angles. The attenuation factors were calculated by counting the gamma rays penetrating a known mass and mixture of pellets. A bracket was placed beneath the duct to hold the pellets in position between the source and the detector, and the attenuation of the gamma source was determined. For these tests, the packing factor of the pellets was assumed to be 0.60. The average number of counts per channel without any pellets was 7505, while the average number of counts with steel pellets was 141.5. The background counting rate was determined to be 59.4. Using Eq. (A.1) the attenuation factors for all steel, all ceramic, and a 50/50 mixture determined to be:

$$\lambda_s = 7.51, \quad \lambda_c = 4.51, \quad \lambda_{50/50} = 6.34 . \quad (A.2)$$

the raw calibration data is shown in Fig. A-2.

Fig. A-2. Summary of calibration data on August 26, 1981.

The next step in calibration was to calculate the packing factor of a known pellet mixture flowing from the duct. For all steel pellets the average count rate was determined to be 1472 for the 45-deg slot. Using Eq. (A.1) along with the calculated attenuation factor, λ_s , results in a calculated packing factor of 0.185. The average packing factor for both the 45-deg and 63-deg slots with either ceramic or steel pellets is calculated to be 0.180. Similar tests were conducted for a 50-50 mixture and for an all-ceramic composition, and the raw data for these tests are shown in Fig. A-3. These data were used to establish a set of calibration curves, shown in Fig. A-4 relating the average counts per channel to the composition of the pellet mixture. The highest count rate was shown for the 33-deg exit angle, indicating that the shallowest angle resulted in the lowest discharge rate from the duct. The discharge rates for the 45-deg and 63-deg cases are approximately equal.

Fig. A-3. Flowing calibration data. Exit angle = 45 deg.

The curves relating the composition of the discharge to the count rate was then input into the computer code so that the relative percentage of steel pellets flowing past the detector could be calculated from the raw data.

Fig. A-4. Calibration data relating average counts to mixture.

A.3.2 Pellet Flow Tests

Typical raw data from pellet flow tests are shown in Figs. A-5 and A-6. Data collection was initiated approximately 5 s (100 channels at 50 ms each) prior to starting the pellet discharge. When discharge begins, the count rate falls due to the presence of the pellets in the gamma-ray beam. The point at which the count rate is the lowest corresponds to the point of highest steel pellet concentration. The count rate then returns to a level indicative of 100% ceramic pellets. The total discharge time is approximately 12 s. The count rate returns to the unattenuated value when the flow stops. The raw data is finally reduced to show the percentage of steel pellets in the gamma-ray beams as a function of time (i.e. channel). The cases which were tested are shown in Fig. A-7. The steel tracer pellets were located between 12.1 cm and 21.6 cm from the exit slot and they were placed in either the front half or back half of the duct. Three exit angles, 33, 45, and 63 deg were tested.

Fig. A-5. Raw data - back half steel from 4.75 to 8.5 in.

Fig. A-6. Raw data - front half steel from 7.75 to 12 in.

The reduced data for the test are shown in Fig. A-5 and A-6 and are plotted in Figs. A-8 and A-9. In general, the percentage of steel pellets reaches a peak at ~ 1 s after flow begins. The spread of the peaks indicates that the steel pellets have mixed to some extent with the ceramic pellets. The curves also show a significant amount of scatter in the data, indicating that several test runs for each set of conditions would be necessary to improve the counting statistics. The combined results of six tests are shown in Figs. A-10 and A-11. Fig. A-10 shows the steel pellet peaks for the tests

Fig. A-7. Test configurations used for flowing pebble tests.

Fig. A-8. Calibrated data, Test No. 10.

Fig. A-9. Calibrated data, Test No. 2.

Fig. A-10. Effect of exit angle. Steel is above exit.

Fig. A-11. Effect of exit angle. Steel is not over exit.

where the steel pellets were placed directly over the exit slot. The figure shows a sharp peak for the 33-deg slot and a more diffuse peak for the 63-deg slot. These results indicate that, in the front of the duct, less mixing occurs in the case of the 33-deg slot than in the case of the 63-deg slot. Conversely, pellets at the rear of the duct move more slowly and show greater mixing than the pellets at the front duct (see Fig. A-11). The 33-deg slot results in the slowest velocity at the rear of the duct and the greatest mixing.

A.4 Conclusions

(1) A technique for quantitatively studying the flow characteristics of a pebble bed has been devised and constructed.

(2) Initial tests show that the relative flow of pellets from the front to the rear of the duct can be controlled by modifying the angle of the exist slot.

(3) The test results confirm the viability of the offset exit concept for controlling the flow rates of pebbles within the duct. A shallow exit angle is preferred to control the velocity of the flow rates from the front to the rear of the duct. Additional tests are required to determine the optimum exit configuration.

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