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Projekt Schneller Brüter

Integrated Design and Optimization of Sodium/Sodium Heat Exchangers with Computer Assistance

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Abstract

The SINEX code allows the design of sodium/sodium intermediate heat exchangers for the steady-state reference operating. The novel feature of the program system consists in the combination of thermal and mechanical calculations. A graphical output has been incorporated. A simplified cross section of each heat exchanger calculated can be plotted. In parameter calculations the influence of the different parameters can also be plotted automatically in diagrams. As an example of the application of the program system, sodium/sodium intermediate heat exchangers, made out of three different grades of austenitic steel, are compared with each other in terms of cost.

Zusammenfassung

Das Programmsystem SINEX erlaubt die Auslegung von Natrium/Natrium Zwischenwärmetauschern für den stationären Auslegungsfall. Die Neuartigkeit des Programmsystems besteht darin, daß die Berechnungsmethoden der thermischen Auslegung mit den Methoden der festigkeitsmäßigen Auslegung kombiniert wurden. Das System macht von den Möglichkeiten der graphischen Ausgabe Gebrauch. Die errechneten Abmessungen eines Wärmetauschers können direkt von der Maschine in eine einfach maßstäbliche Zeichnung umgesetzt werden. Daneben lassen sich die verschiedensten Abhängigkeit einzelner Auslegungsgrößen automatisch in Diagrammform darstellen. Um die Fähigkeiten des Systems zu veranschaulichen, werden in einem Beispiel Natrium/Natrium-Zwischenwärmetauscher aus drei verschiedenen austenitischen Stählen wirtschaftlich miteinander verglichen.

INTERNATIONAL CENTRE FOR HEAT AND MASS TRANSFER

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INTEGRATED DESIGN AND OPTIMIZATION OF SODIUM/SODIUM HEAT EXCHANGERS WITH COMPUTER ASSISTANCE

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Introduction

The heat exchangers in the primary system of a reactor facility are among the components which have a decisive influence on availability, safety and economy of a nuclear power station.
Safety and economy are requirements not easily made compatible.
It is a matter of fulfilling, in an economically bearable way, the sometimes very stringent safety provisions imposed upon the plant by licensing authorities. This requires special methods of selection allowing the optimum solution to be found for each specific case from among a large number of possible variants. However, from experience it is known that these methods require considerable

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computational efforts which, usually, can be coped with only by means of electronic data processing systems, provided that adequate program systems are available.

The SINEX /1/ program system (sodium intermediate heat exchanger) is such a program system. It allows the design of sodium/sodium intermediate heat exchangers for steady-state design conditions. The novel feature of the program consists in the combination of, mostly,standard methods of calculation for the thermal and mechanical design. As far as the heat exchanger is concerned, this means a thermal design of the tube bundle and a simple mechanical design of the most important heat exchanger components.
The relatively simple input and the clear and easily interpreted output make the system suitable also for parameter studies and optimization calculations. In addition to the usual print output the results calculated may be plotted. A simplified cross section of each heat exchanger calculated can be plotted. This drawing will also list the main design parameters. In addition, the temperature curves of the coolant or the heating surface can be plotted in diagrams versus power and the temperature curves can be plotted versus the heating surface.

In parameter calculations the influence of the different parameters can also be plotted automatically in diagrams by the machine.

The high flexibility of the modular program system will easily allow an extension to other types, different coolants or other methods of calculation. This will ensure the possibility of adapting the system to individual requirements in many cases.

Description of Heat Exchanger

The type presently implemented is shown in Fig. 1. The heat exchanger has a straight or helical tube bundle, operates by the counter flow principle and is variable with respect to the flow paths. The primary, that is, the hotter coolant may flow both on the tube side or on the shell side of the exchanger tubes. In this way, the natural free convection flow is retained as much as possible. The tube bundle is located between two tube plates. It is penetrated by a multilayered central tube and surrounded by a flow skirt on the outside. The top tube plate is clamped between the vessel flanges. The bottom tube plate is part of the lower plenum construction which is movable in the vertical direction and thus can follow the thermal expansions of the tube bundle.

Description of Program System

The schematic diagram of the program system is shown in Fig. 2. The main control program controls the program sequence, assigns standard values to the input data and stores the data in a temporary file after each individual computation step.

The remaining section of the system can be subdivided into four main parts:

- In the first part, the tube bundle is geometrically subdivided and the thermal design calculation is performed. This is followed by the calculation of the pressure drop.

- The second part includes a material properties library, which contains, as a function of temperatures, all the thermodynamic data required for the calculation, the material data and characteristics, both for the coolant and for the steel grades used.
- In the third part, the mechanical design of the vessel, the tube plates and flanges, and of the other major components is carried out. This part of the system is responsible also for the print output of the calculated results.
- In the fourth part, the data are processed for the plot output. For this step, also information from the temporary storage is available.

The main parts outlined above are composed of a number of sub-programs in which partial calculations are performed.

Method of Calculation and Relationships

The Nusselt-correlations determining the heat transfer are shown in Fig. 3.

For the tube bundle with cross flow Eq. (1) applies according to RICKARD /2/. For the tube bundle with parallel flow, the Nusselt-number is calculated according to DWYER /3/, Eq. (2), and FRIEDLAND and BONILLA /2/ Eq. 3, respectively. For the flow in circular tubes the relationships of SEBAN, SHIMAZAKI /4/, Eq.(4) and DWYER /5/, Eq. (5) hold respectively.

The shell side pressure drop for the cross flow tube bundle is calculated according to GRIMISON /6/. If the flow is directed diagonally to the tube bundle the influence according to WIEMER /7/ is taken into account. In tube bundles with parallel flow the pressure drop is determined in accordance with MOODY /8/ on the basis of studies by VON KARMAN and NIKURADSE, the friction factor requiring correction in accordance with KAUL and VON KISS /9/. On the tube side, the applicable relationships are those of PRANDTL and VON KARMAN and COLBROOK and WHITE /10/, respectively. If the tubes are curved, the friction factor is corrected in accordance with HAUSEN /11/.

In baffled tube bundles the shell-side pressure drop is calculated according to BELL /12/, which supplies the best results compared with other methods /13/.

For the mechanical design of the heat exchangers some basic assumptions must be made. It is assumed that the diameter of the vessel is equal to the diameter of the flange. The diameter of the central tube is determined on the basis of a given flow velocity. The diameter of the bottom plenum is assumed to be equal to the tube bundle diameter. The wall thickness of important components is calculated in accordance with AD data sheets /14/. The calculated values are limited at the lower end by a minimum wall thickness given by manufacturing considerations.

The tube plates are designed in accordance with O'DONNEL and LANGER /15/ by the method of equivalent elastic constants. The perforated plate in this case must be calculated like an equivalent unperforated plate by the familiar relationships, but with the elastic constants properly corrected.

The flange design is based on work by SCHWAIGERER /16/ and on the tentative DIN 2505 standard, respectively /17/. The method does not allow the flange dimensions to be calculated directly. Hence, they are iterated until the calculated stresses in the flange are within the permissible range. The permissible stress has been reached if the following conditions have been fulfilled:

Simple safety relative to the yield point σ_S or the 0.2% yield strength $\sigma_{0.2}$, 1.5fold safety relative to the creep rupture strength $\sigma_{B/100\ 000}$, and simple safety relative to the time yield limit $\sigma_{1/100\ 000}$. In this case, the three modes of operation to be taken into account are conditions of installation, test condition, and operating condition.

Examples of Application

As an example of the application of the program system, sodium/ sodium intermediate heat exchangers made of three different grades of austenitic steel are compared with each other in terms of cost. The design causing the lowest costs is to be found. The different materials are given in Fig. 4. The individual variants were assumed to be made completely of the same material. The study takes into account not only the different material prices and costs of fabrication but also the different material characteristics.

The determination of the price is based on specific prices which are taken into account in DM/m^2 for the raw bundle with internals and tube plates and in DM/kg for the vessel including flanges. With respect to these prices, it was assumed that all the costs of materials, fabrication and testing were included, all fabrication problems had been solved and that there were no increases in costs due to costs of development.

The assumed cost per unit heating surface for the tube bundle is shown in Fig. 5 as a function of the tube outside diameter with the tube wall thickness as a parameter. It should be added that no information is given here about the absolute level of prices.

The trend of the curves and their relation to each other is likely to come close to the true curve for heat exchangers of the size studied.

The assumed cost per unit mass for the vessel is given in Fig. 6. In most cases, the operating conditions are given from the overall plant. Hence, they were initially kept constant in the example on the basis of the assumptions shown in Fig. 7.

Results

The outside diameter of the exchanger tubes was varied in the range between 12 and 36 mm with wall thicknesses of 2.0, 1.5 and 1.0 mm. The results are shown in Fig. 8. Also the overall costs are plotted over the outside tube diameter, the parameters being the material and the tube wall thickness.

The dash-dotted curves apply to 4436 grade steel, the dashed ones to 4948 grade steel, and the solid lines to 4961 grade steel.

The highest cost is obtained for 4436 grade steel, which is also characterized by the highest specific prices. Although the specific prices of 4948 steel are lower than those of 4961 steel, the capital costs turn out to be higher. This is due to two reasons in this case. On the one hand, 4961 steel has a higher thermal conductivity, on the other hand,it also has a higher strength.

It should be noted that there is a significant influence of the wall thickness of the tube upon the overall cost. This may be explained by the fact that the heat resistance in the tube wall is 2.5 to 5 times higher as compared to the boundary layer on the inner and outer sides of the tube.

Fig. 9 shows three variants made of different grades of steel for which the costs turn out to be a minimum; they are compared in basic diagrams true to scale. The lowest capital costs are shown by the design made of 4961 steel shown in the center with dimensions of the exchanger tubes of 18 x 1 mm. Although this variant has a greater structural volume than the two others, it is less expensive at the specific prices assumed. The other two designs have dimensions of the exchanger tubes of 16 x 1 mm and approximately the same structural volume; however, the 4948 design is the less expensive one.

Fig. 10 shows the heating surface plotted versus the outside tube diameter. Again, the parameters are the material and the wall thickness of the tube. An incrase in the wall thickness of the tube by 0.5 mm leads to an increase in the heating surface by approximately 1 m^2/MW.

Fig. 11 shows the influence of the outside tube diameter on the structural volume. Again, the parameters are the material and the wall thickness of the tube. An increase in wall thickness brings about an increase in construction volume by roughly 0.015 m^3/MW.

These examples referred to constant operating data. Since the program system takes into account all the functional relationships concerning the heat exchanger, answering also any other question is a mere matter of routine. The following figures are examples of variations of the operating data. The influence of the pressure drop on the overall cost and the structural volume is investigated for the minimum cost variants.

Fig. 12 shows the influence of the primary pressure drop on the overall costs with the material as a parameter. The curve behaves as one would expect. An increase in the primary pressure drop results in a decrease of the capital costs. The same trend is reflected also in the influence upon the structural volume in accordance with Fig. 13.

In Fig. 14 the influence of the secondary pressure drop on the overall cost is shown. Here, the trend is reversed. An increase in secondary pressure drop does improve the heat transfer on the tubes side, but also increases the tube length. Since the primary pressure drop remains constant, heat transfer on the shell side will thus be reduced. The deterioration on the shell side exceeds the improvement on the tube side. Hence, there is an increase in the heating surface. Another increase in costs arises from the increase in the height of the vessel. This also exceeds the slight reduction of the diameter of the vessel. In this way, one can explain why an increase in the secondary pressure drop results in an increase in the overall costs.

Fig. 15 represents the influence of the secondary pressure drop on the structural volume. Again, the effect is as described above. An increase in pressure drop will increase the structural volume.

These examples show the tremendous advantages and the possibilities inherent in the use of EDP systems in engineering design. The application of such program systems has made it possible to increase the productivity of the planning engineer in a way never experienced before. Since all the routine work, which is such an expense of time, is thus kept away from man, man has found time again to devote his attention to the broad outlines of a higher order. This allows him to pursue many problems that would have gone unanswered in former times, also because of the tremendous amount of expenditure involved in their solution.

The program system outlined above does not yet supply a detailed design of the heat exchanger ready for fabrication. However, it does make available all the information which is necessary for fundamental decisions and analyses. The concept selected hardly imposes any limits on the extensibility of the system. Hence, it will be possible in most cases to adapt it to specific situations and conditions.

Nomenclature

d_a	Outer tube diameter
Nu	Nusselt number
Pe	Peclet number
Pr	Prandtl number
t	Pitch of tube
ε_M	Eddy diffusivity for momentum transfer
γ	Kinematic viscosity
ψ	Average value of eddy diffusivity for heat transfer to ε_M

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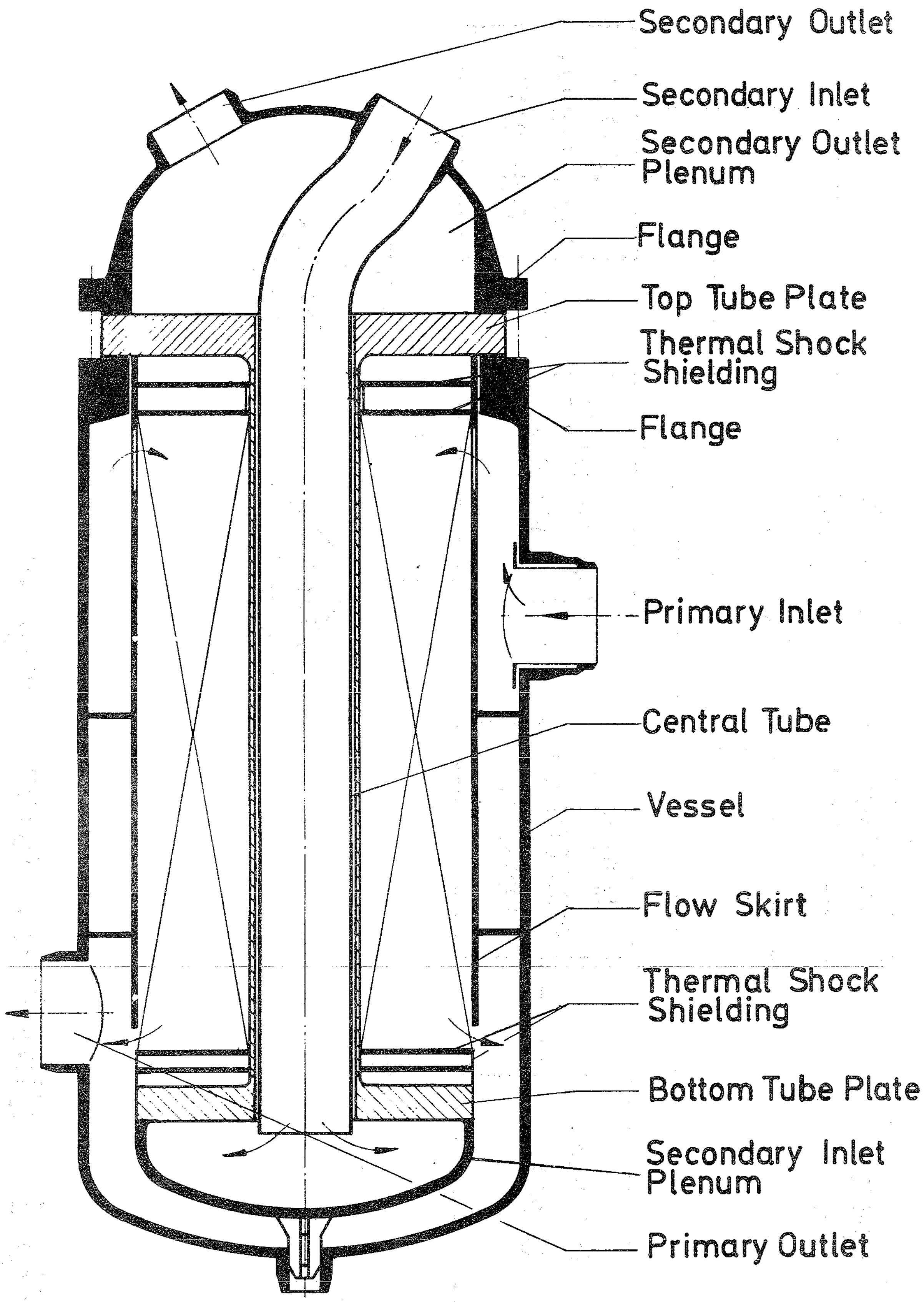

Fig. 1 Sodium/Sodium Intermediate Heat Exchanger Cross Section

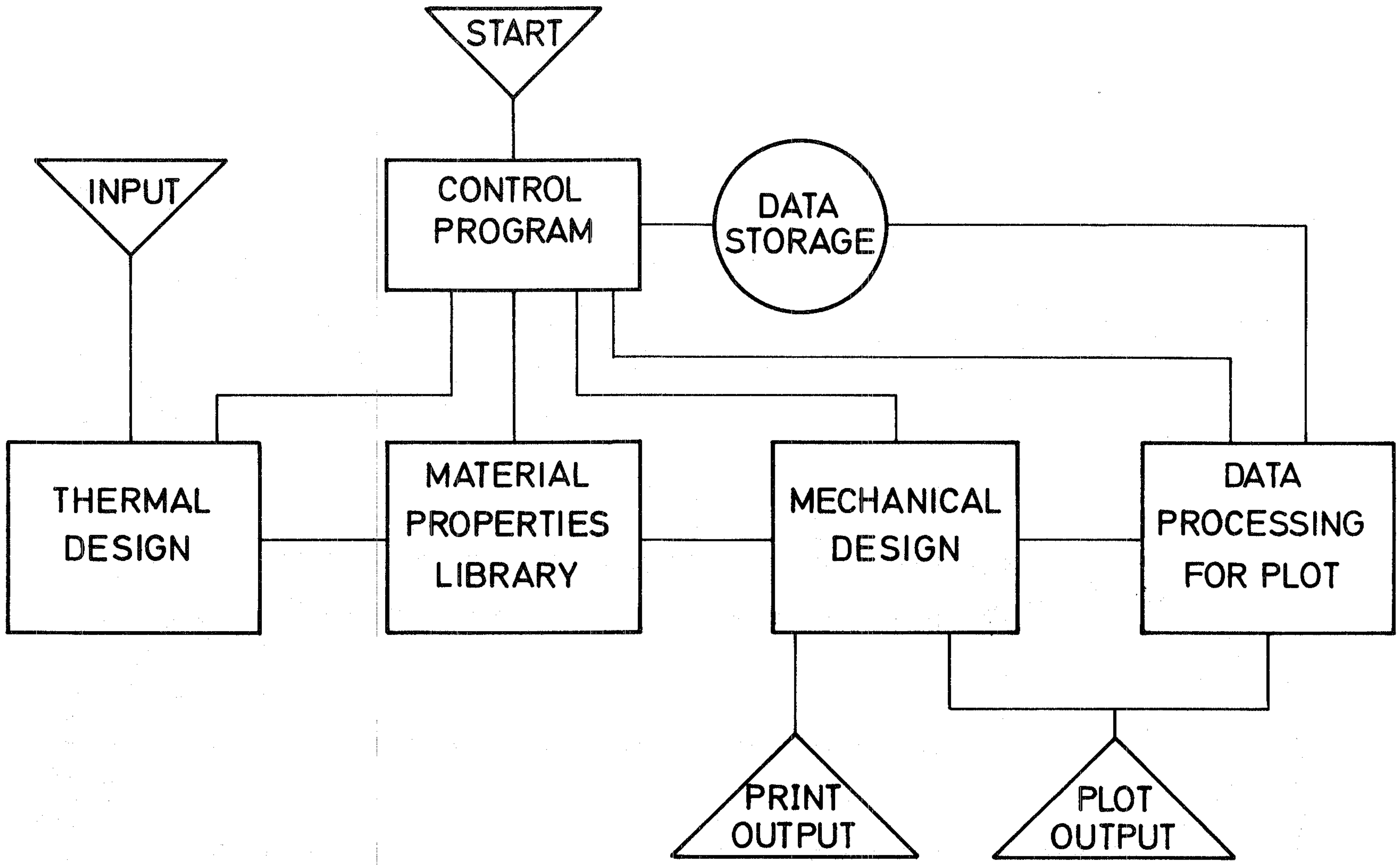

Fig. 2 Simplified Block Diagram

$\underline{50 < Pe < 4000}$ (1)

$$Nu = Pe^{0.5}$$

$\underline{100 \leq Pe \leq 10\,000\, ; \quad 1.3 < \frac{t}{da} \leq 2.5}$

$$Nu = 0.93 + 10.81 \left(\frac{t}{da}\right) - 2.01 \left(\frac{t}{da}\right)^2 + 0.0252 \left(\frac{t}{da}\right)^{0.273} (\Psi \cdot Pe)^{0.8} \quad (2)$$

$$\Psi = 1 - \frac{1.82}{P_r (\varepsilon_M / \nu)^{1.4}_{max}}$$

$\underline{2.5 < \frac{t}{da} \leq 10.0}$ (3)

$$Nu = 7 + 3.8 \left(\frac{t}{da}\right)^{1.52} + 0.027 \left(\frac{t}{da}\right)^{0.27} Pe^{0.8}$$

$\underline{Pe < 200}$ (4)

$$Nu = 5.0 + 0.025\, Pe^{0.8}$$

$\underline{200 \leq Pe \leq 20\,000}$ (5)

$$Nu = 7.0 + 0.025 \left[Pe - \frac{1.82}{(\varepsilon_M / \nu)^{1.4}_{max}} \right]^{0.8}$$

Fig. 3 Nu - Numbers

NO.	DIN-SPECIFICATION	DIN-NO.	AISI-NO.
1	X8 Cr Ni Nb 16 13	(1.4961)	
2	X5 Cr Ni Mo 18 12	(1.4436)	316
3	X6 Cr Ni 18 11	(1.4948)	304

Fig. 4 Austenitic Steels Investigated

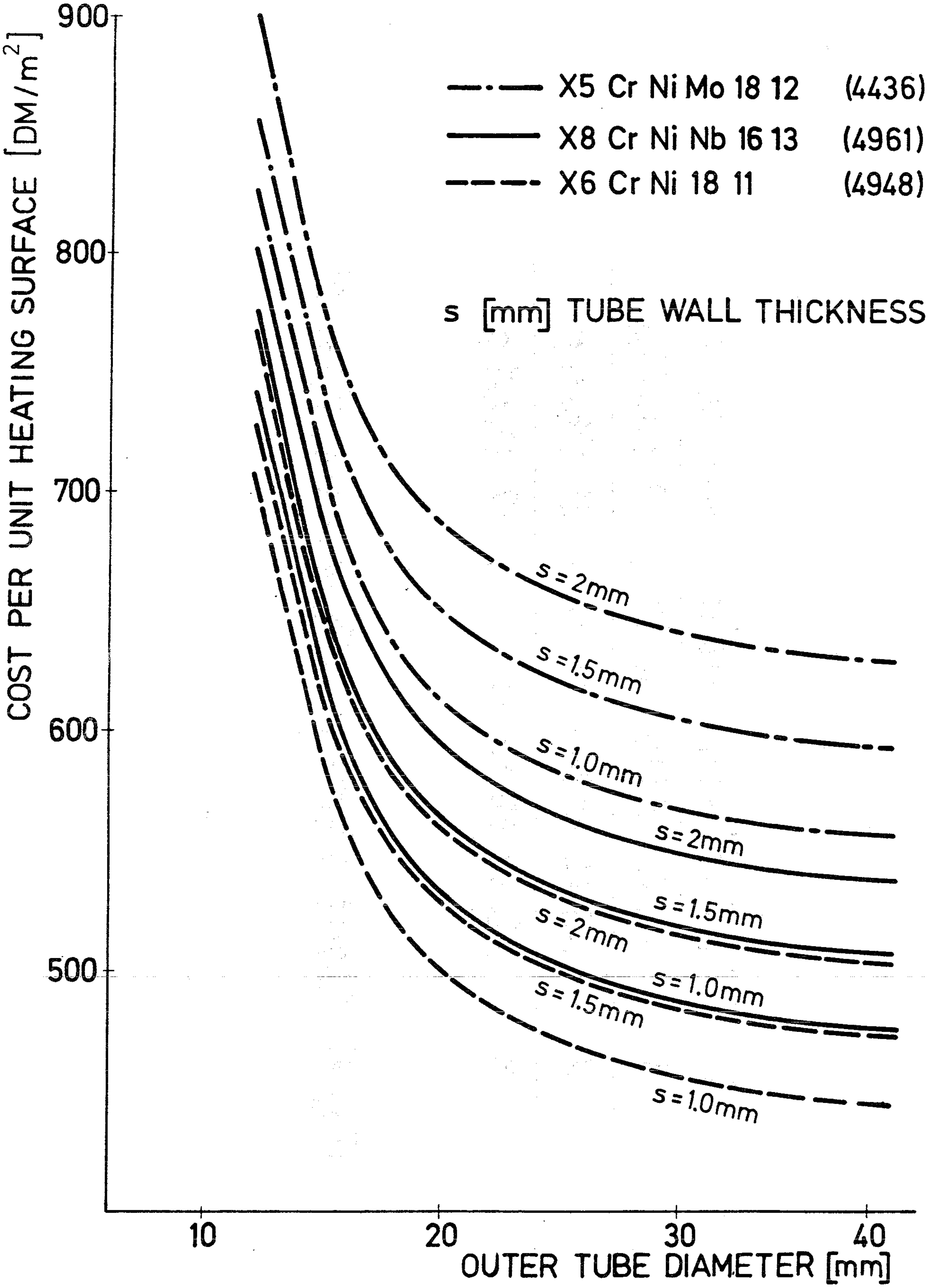

Fig. 5 Cost per Unit Heating Surface Versus Outer Tube Diameter

NO.	DIN - SPECIFICATION	UNIT COST
1	X8 Cr Ni Nb 16 13	18 DM/kg
2	X5 Cr Ni Mo 18 12	21 DM/kg
3	X6 Cr Ni 18 11	17 DM/kg

Fig.6 Cost Per Unit Pressure Vessel

Thermal power	250 MW
Primary inlet temperature	550 °C
Primary outlet temperature	380 °C
Secondary inlet temperature	340 °C
Secondary outlet temperature	530 °C
Primary pressure drop	0.7 at
Secondary pressure drop	0.7 at
Primary design pressure	10 at
Secondary design pressure	15 at

Fig. 7 Reference Operating and Design Data

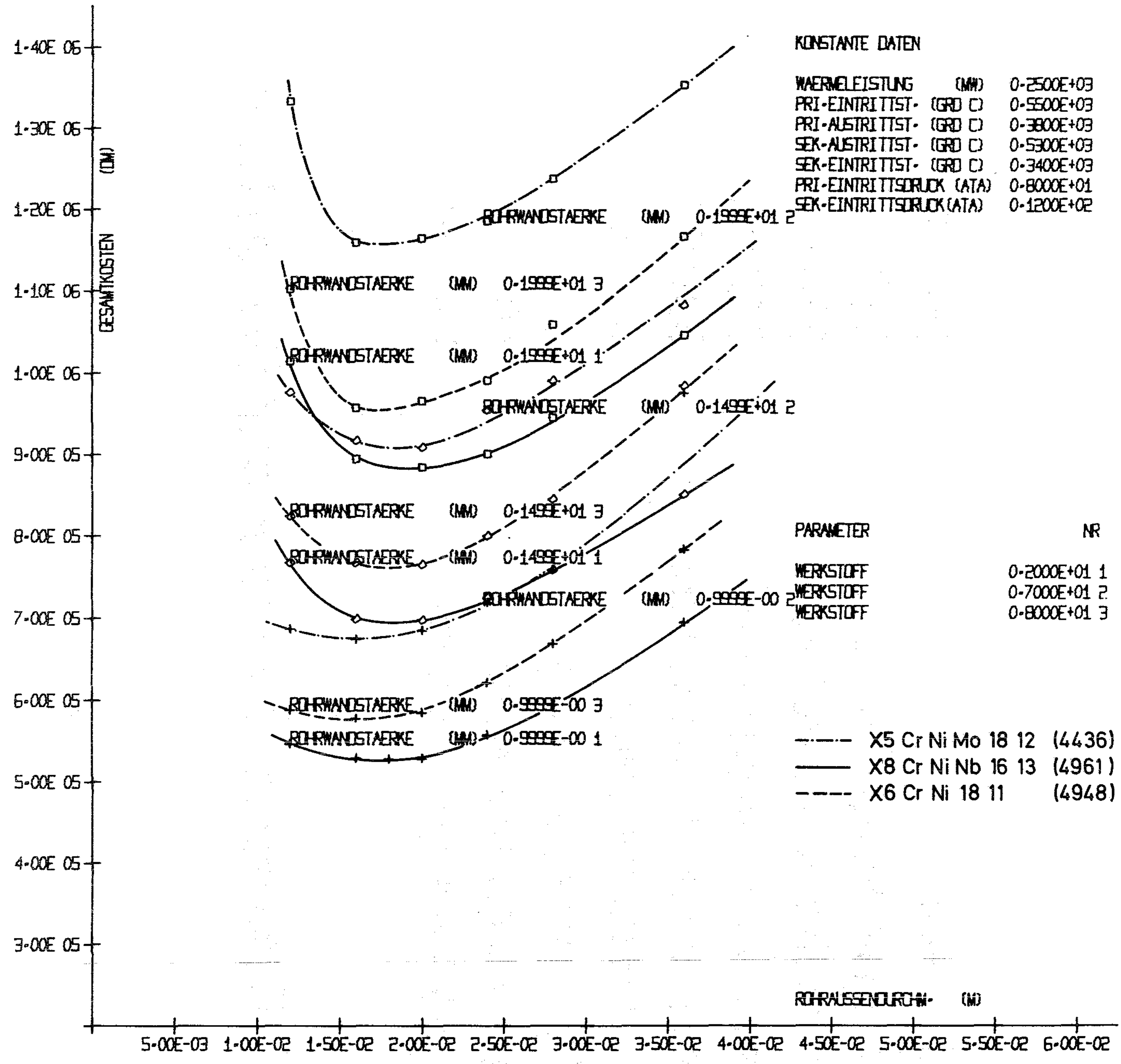

Fig. 8 Total Heat Exchanger Costs Versus Tube Outer Diameter

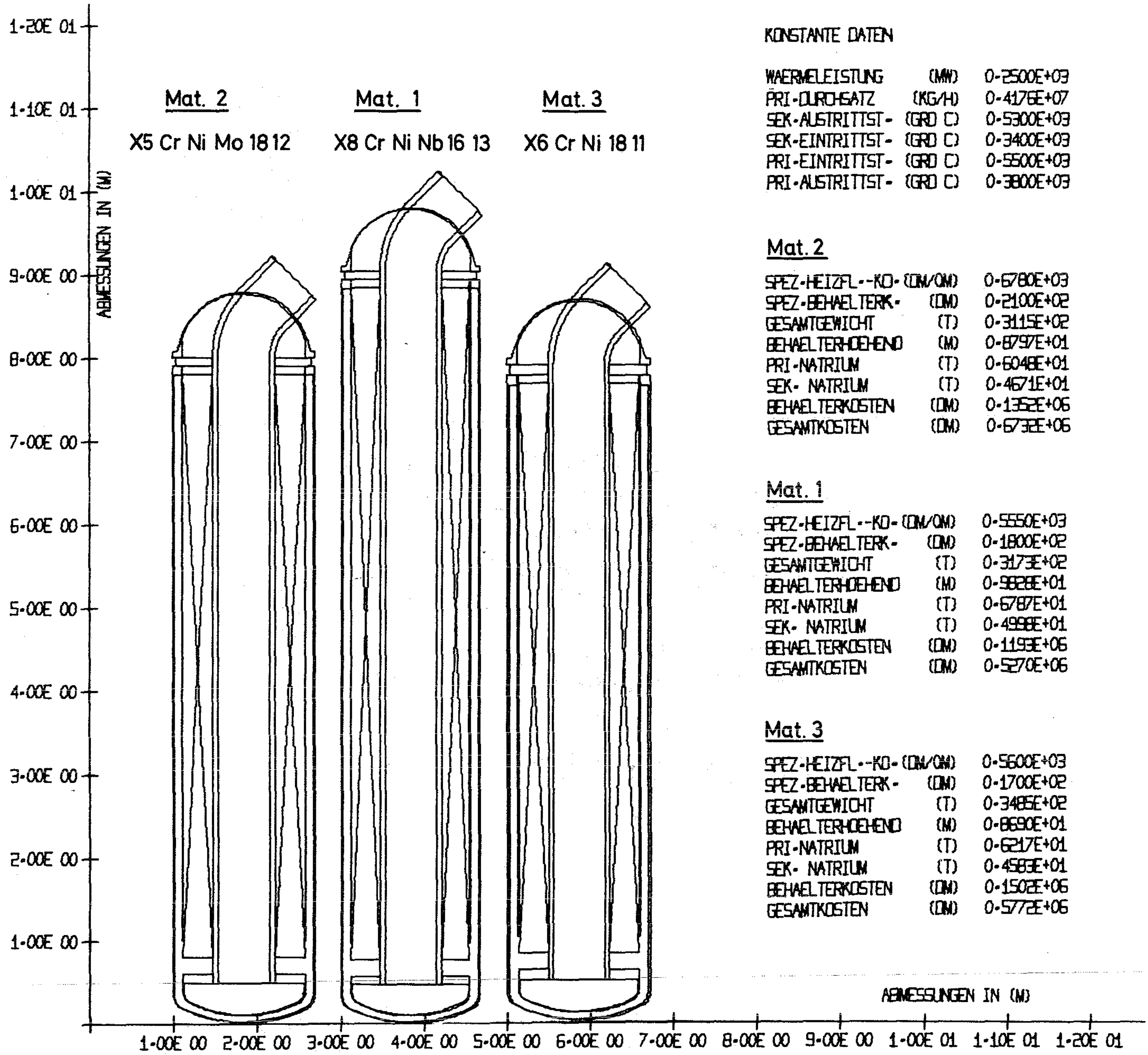

Fig. 9 Comparison of Optimum Heat Exchanger Design for three Different Austenitic Materials

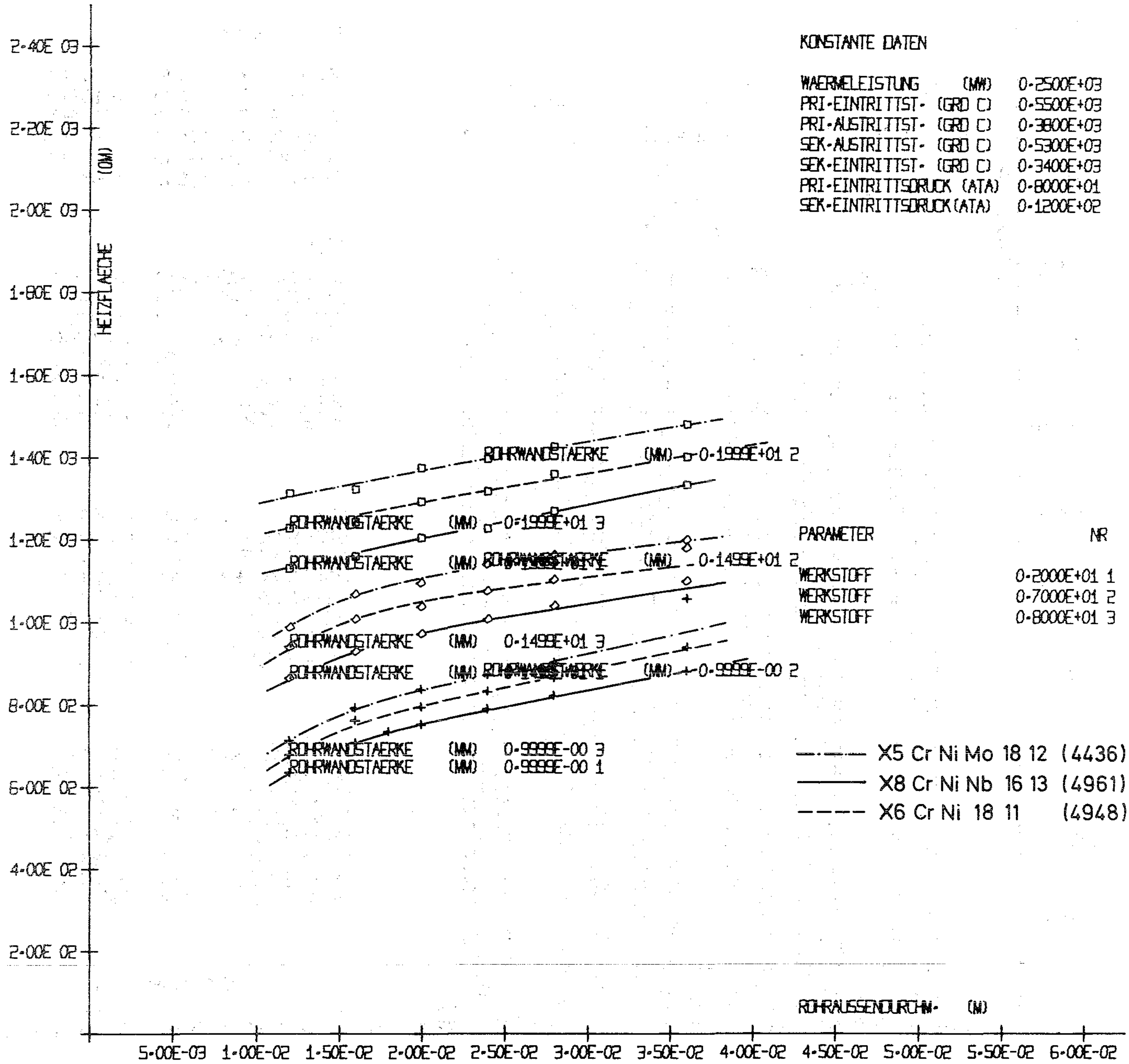

Fig. 10 Heating Surface Versus Outer Tube Diameter

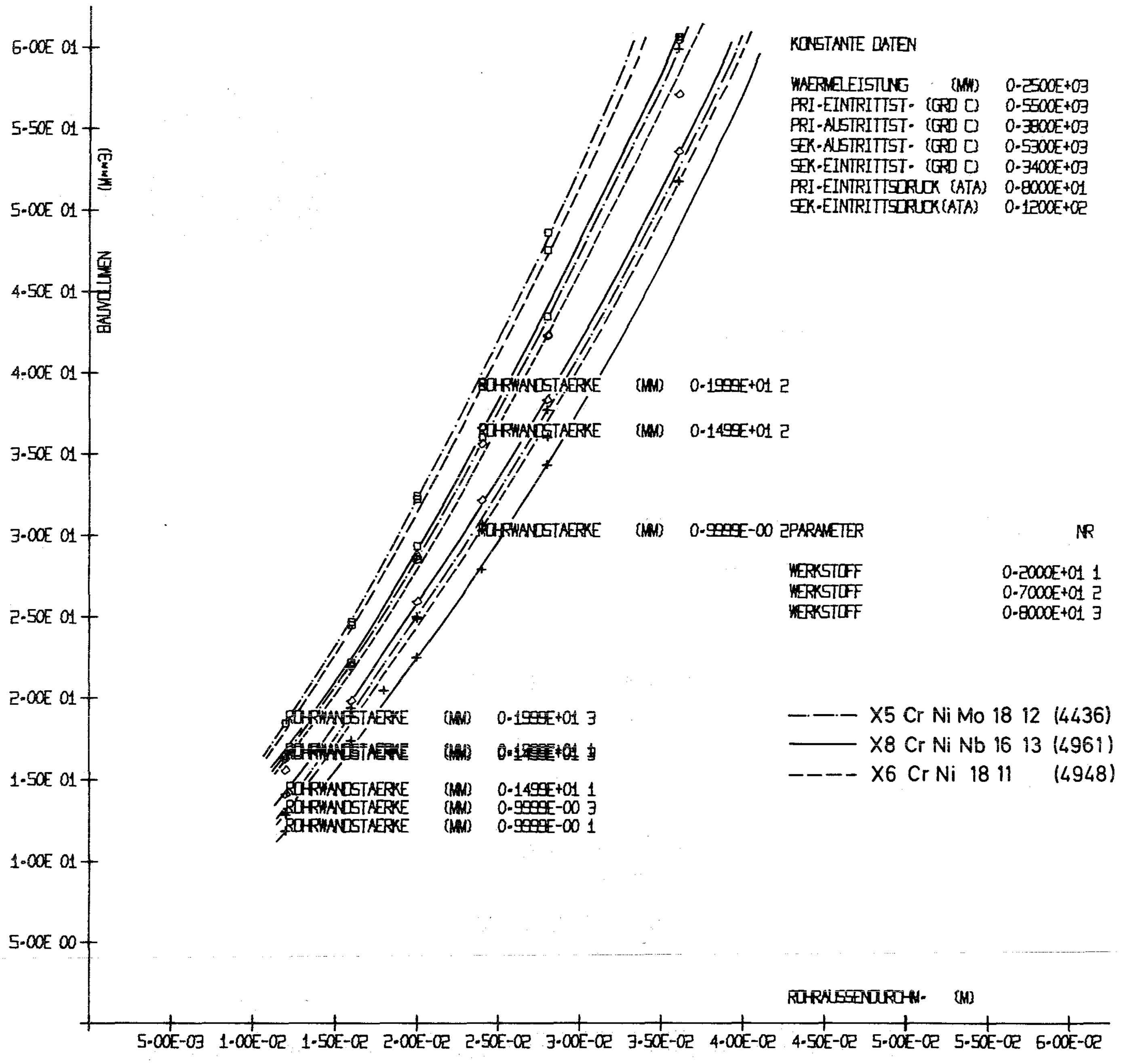

Fig. 11 Total Heat Exchanger Volume Versus Outer Tube Diameter

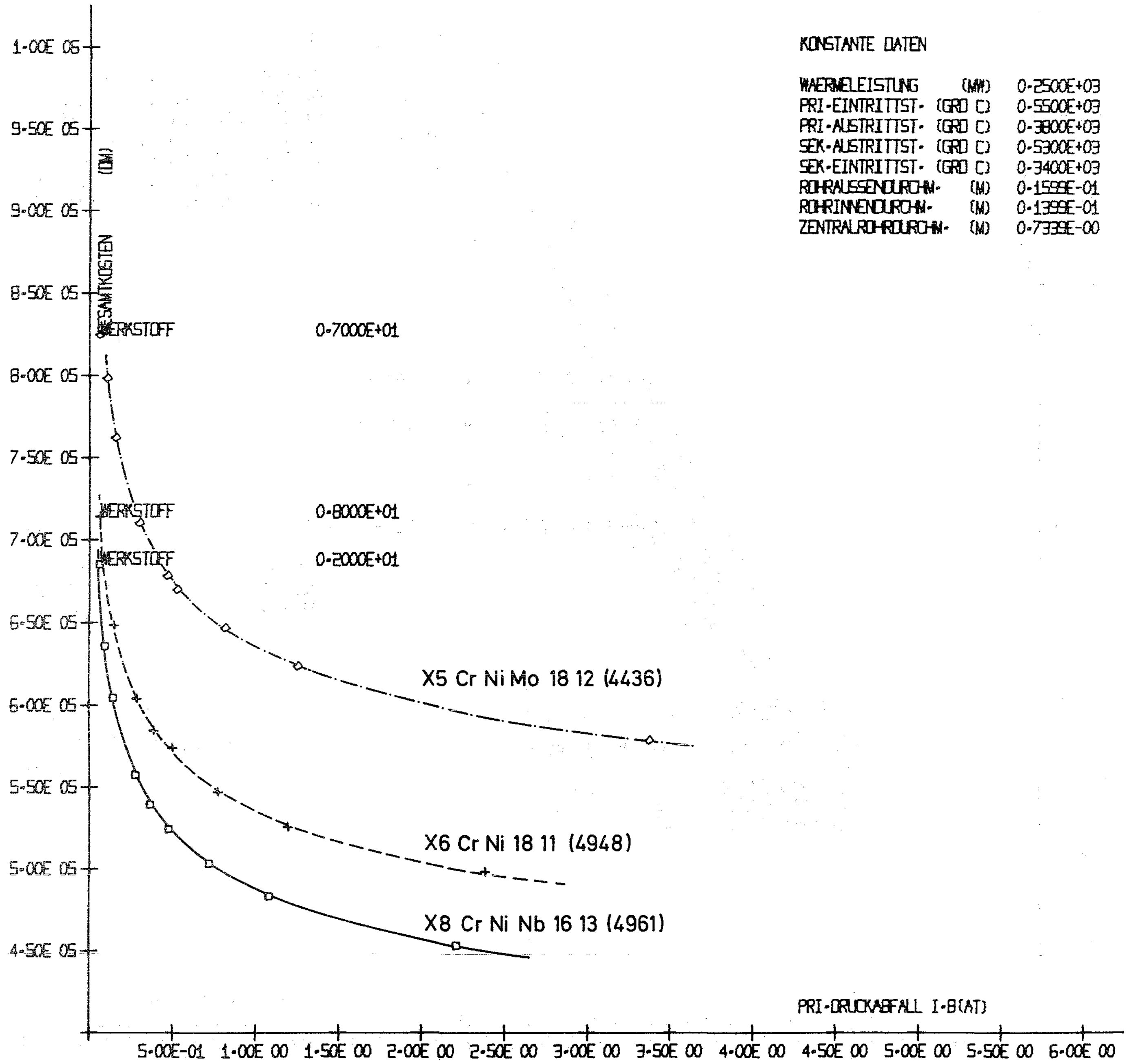

Fig. 12 Total Heat Exchanger Costs Versus Primary Pressure Drop

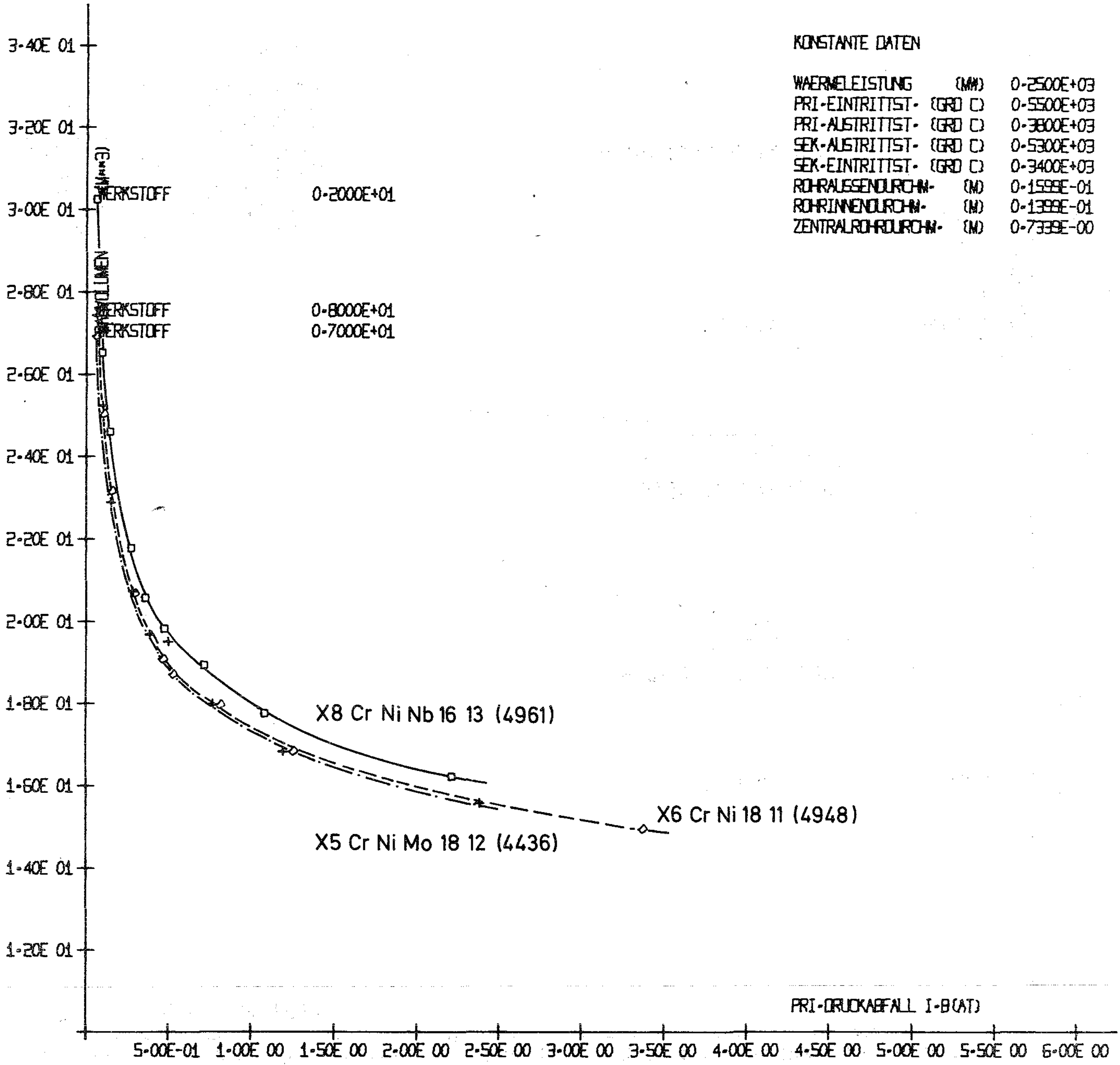

Fig. 13 Total Heat Exchanger Volume Versus Primary Pressure Drop

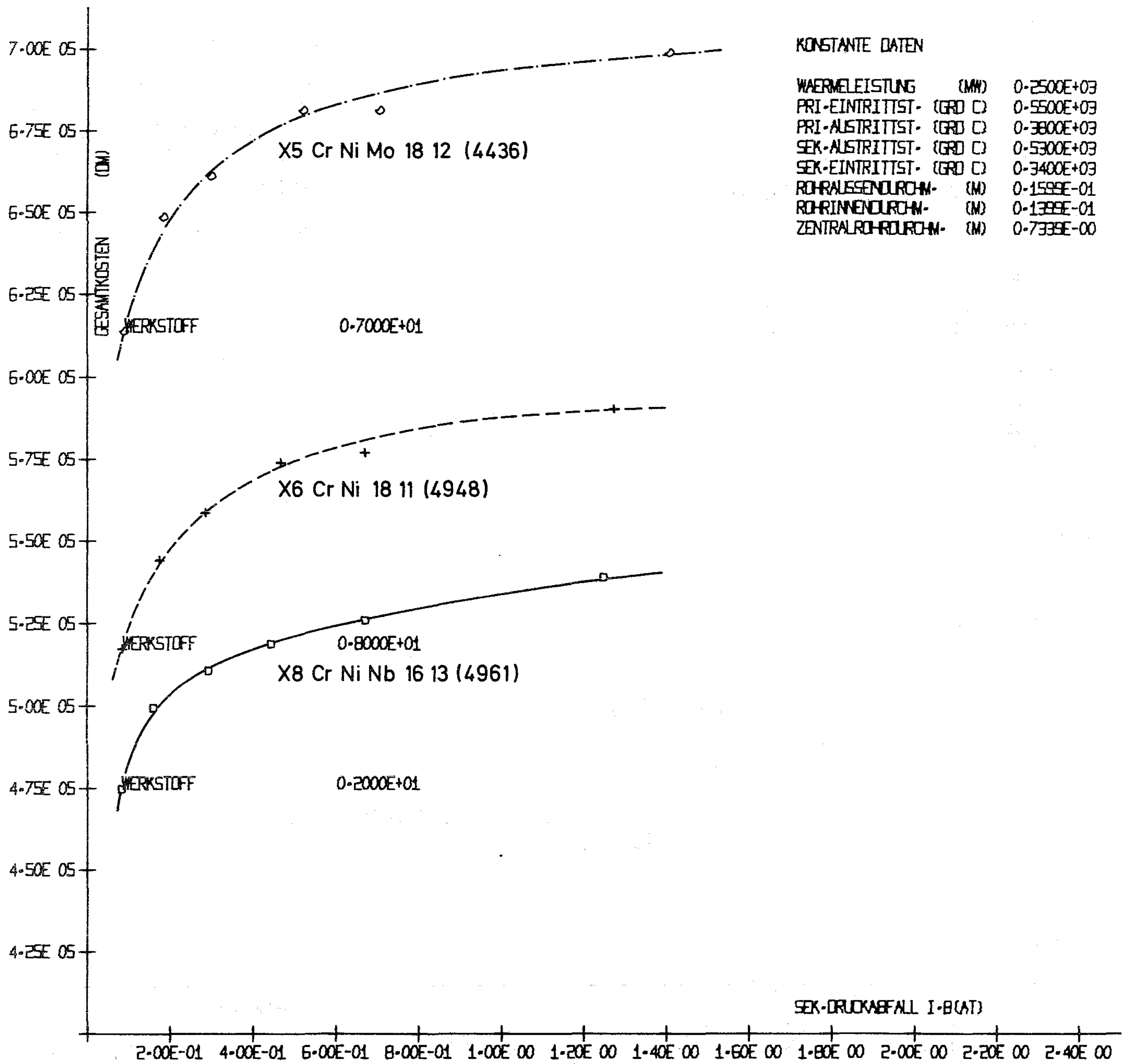

Fig. 14 Total Heat Exchanger Costs Versus Secondary Pressure Drop

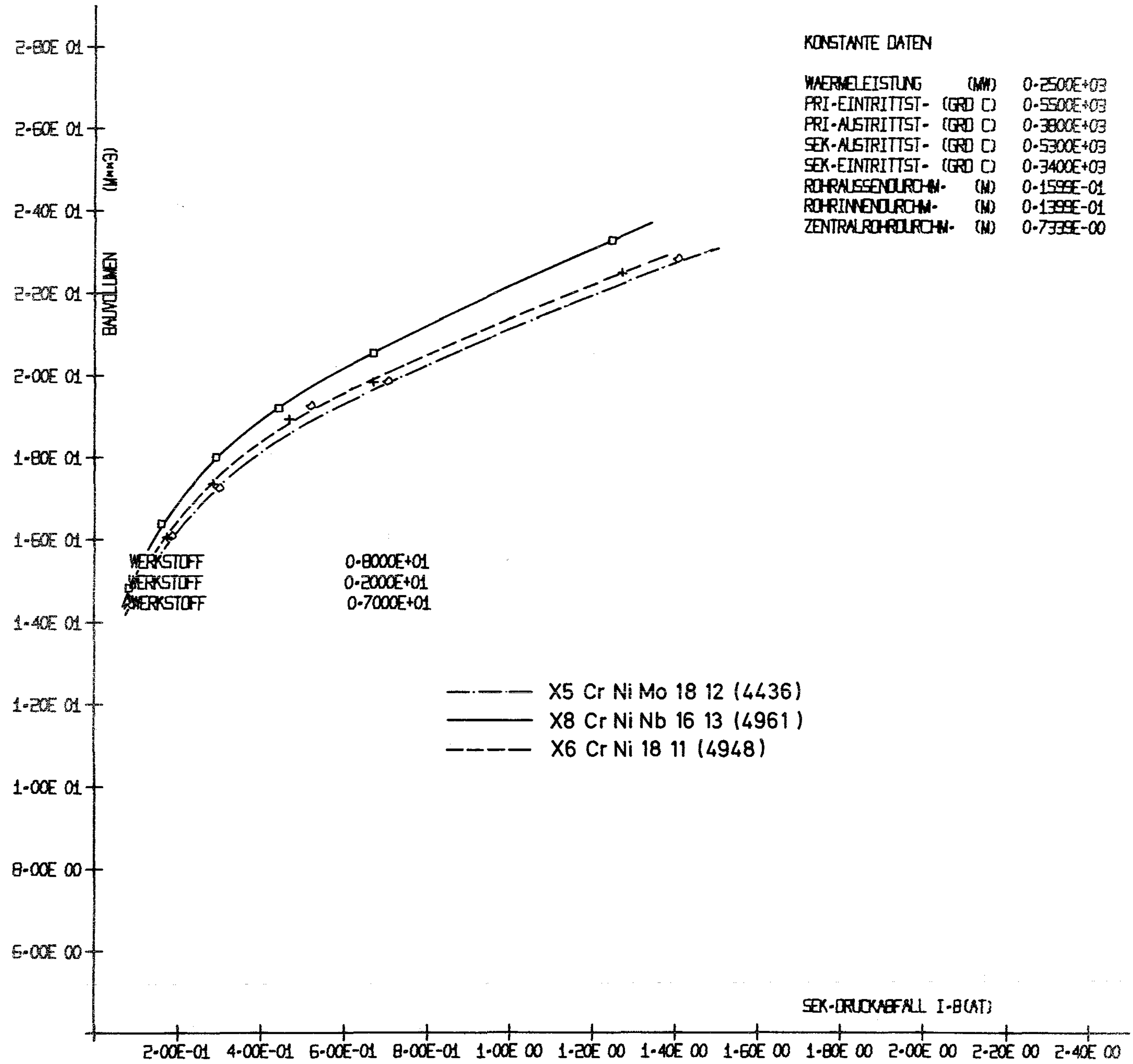

Fig. 15 Total Heat Exchanger Volume Versus Seconday Pressure Drop