Shell and Tube Heat Exchanger Design Comparison

HTRI and Aspen EDR

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HEAT EXCHANGER

A heat exchanger is an equipment used to transfer thermal energy(enthalpy) between two or more fluids which are at different conditions (temperature, pressure, flow). [1]

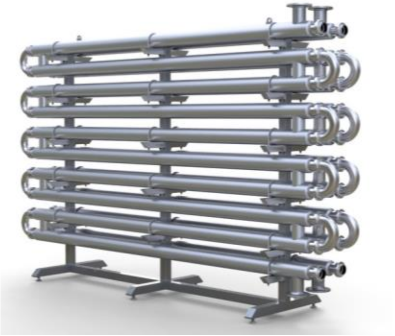
It is widely used for different industry applications (in heating and cooling of process streams) such as: [2]

1. Air conditioning systems.
2. Petrochemical plants
3. Petroleum field.
4. Power plants.
5. Cryogenic plants.
6. Etc.

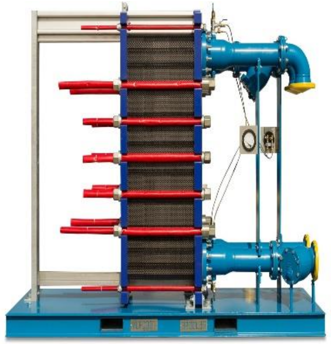
Since heat exchangers are involved in almost all industrial processes, it is highly recommended to know how to design them using different methods: Hand made and Computer Aided Software's.

HEAT EXCHANGER:TYPES

There are a wide variety of heat exchangers. The most used are:



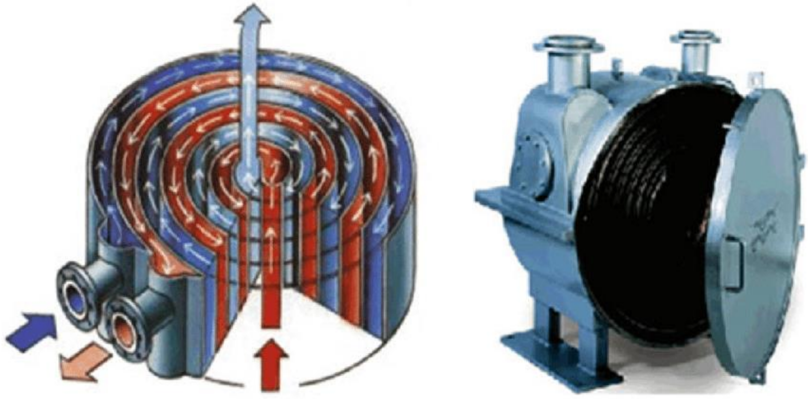
Double Tube



Plate

Air Cooler

Shell & Tube



Spiral Plate

HEAT EXCHANGER

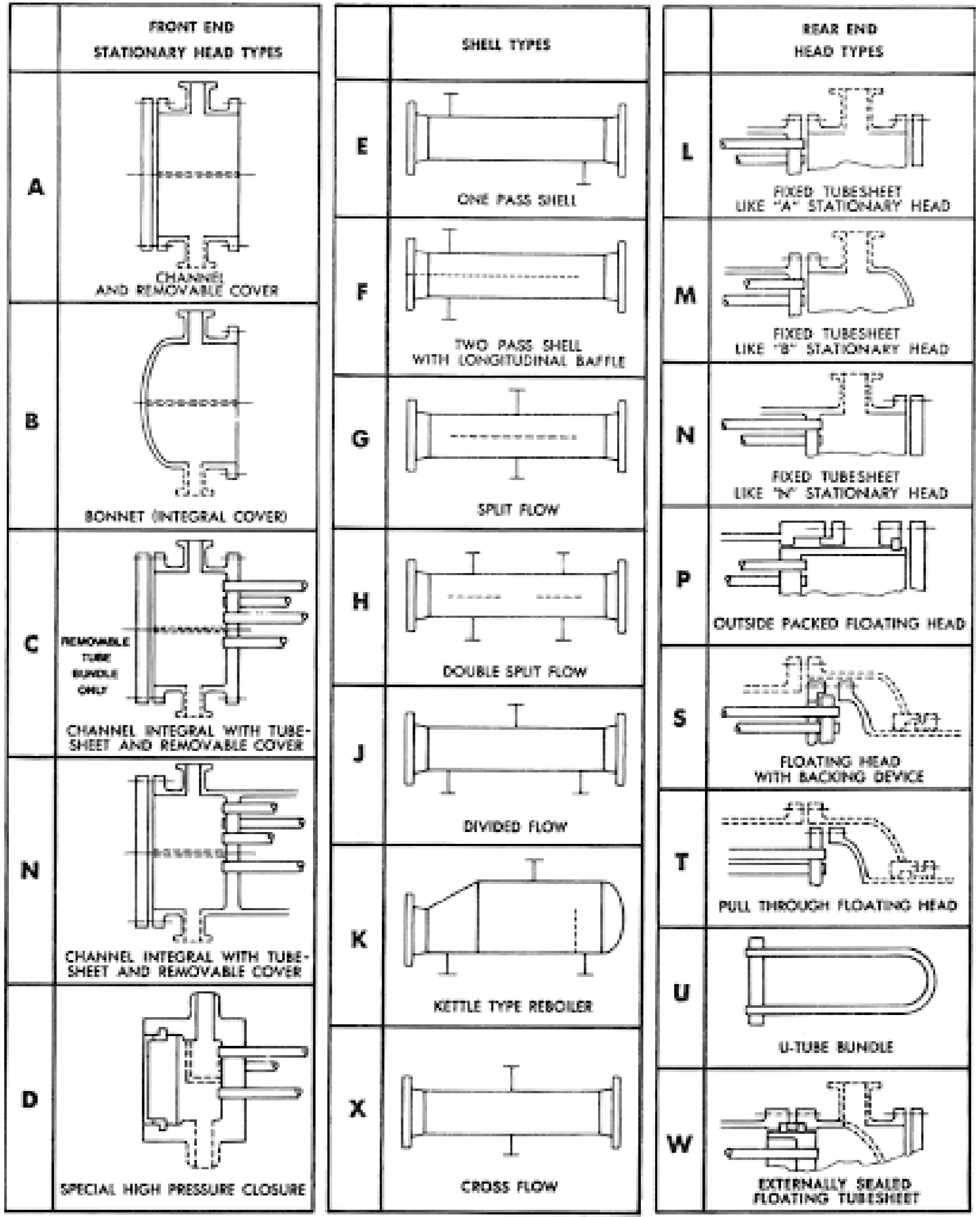
This presentation aims to do a Shell & Tube heat exchanger design comparison using the following methods:

1. Hand Made (Already designed by SERTH) [4]
2. HTRI software
3. Aspen Exchanger Design and Rating (EDR)

Before starting it is important to highlight that all methods used are based under TEMA heat exchanger configurations.

All methods, basically, follows a common algorithm. Nevertheless, it will be noted which one is more practical and reliable at the moment of designing Shell &Tubes Heat Exchanger.

HEAT EXCHANGER: TEMA



[3]

HEAT EXCHANGER:CONSTRAINTS

Before beginning any procedure, it is important to know that heat exchangers must meet two main constraints to be suitable for the service. Therefore, before starting to design, it is firstly more important, knowing how to **evaluate** the constraints.

Thermal evaluation:

Parting from the heat transfer developed: Convection(tube fluid)+Conduction(through pipe thickness)+ Convection(Shell fluid).

$$Q = \left(m\, C_p \Delta T\right)$$

$Q = U_{Req} A\, F\, (\Delta T_{ln})_{cf}$ where $U_{Req} = \frac{Q}{A\, F\, (\Delta T_{ln})_{cf}}$

$Q = U_{clean} A \Delta T_m$ where $U_{clean} = \left(\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o}\right)^{-1}$

U_{clean} is used when the Heat Exchanger is new. While U_D is used when dirt or scale appears.

$Q = U_D A\, F\, (\Delta T_{ln})_{cf}$ $U_D = \left(\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o} + \frac{R_{Di} D_o}{D_i} + R_{Do}\right)^{-1}$

To be thermally suitable, the overall coefficients must:.

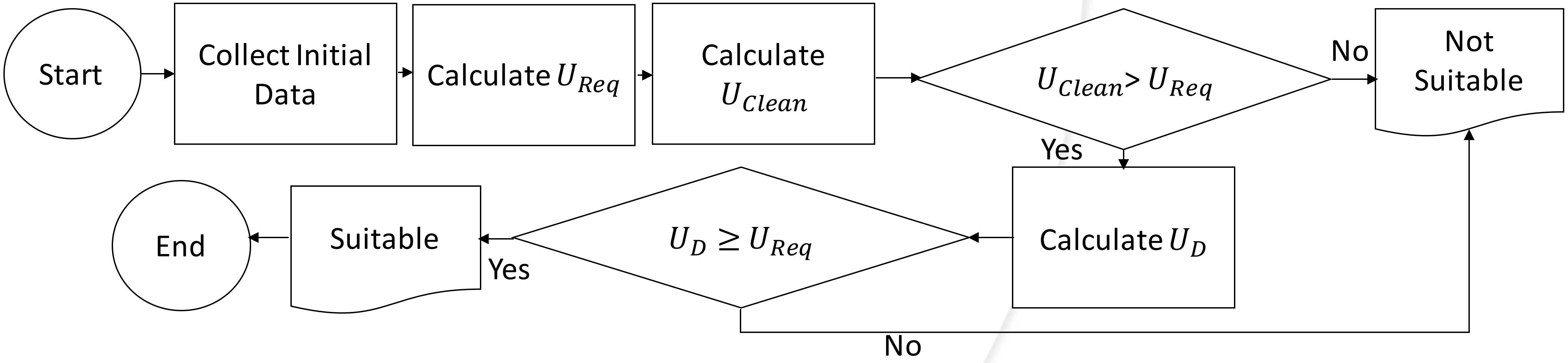
$$\boldsymbol{U_{clean} > U_D > U_{Req}}$$

HEAT EXCHANGER:CONSTRAINTS

Assumptions

- Generally, for having an idea(when exchanger geometry is unknown) about heat transfer preliminary area, it is used a U_D provided by tables (starting point). [4]
- The heat transfer coefficients h_i and h_o are calculated using Nusselt number (for fully developed pipe flow) and Delaware correlations, respectively.
- The literature states that the correction factor (F) should be greater than 0,8. In case F is lower than 0,8 ; it is recommended to increase shell passes. [4]

Thermal Evaluation Diagram Flow



HEAT EXCHANGER:CONSTRAINTS

Hydraulic evaluation: Tube and shell side total pressure drop must be lower than pressure drop allowed.

Once an initial exchanger geometry is chosen, the following equations(When using English Units) are used to check if the initial geometry meets the pressure drops allowances.

Tube Side $\Delta \boldsymbol{P_{tubes}} = \Delta P_f + \Delta P_r + \Delta P_n$

$$\Delta P_f = \frac{f n_p L G^2}{(7{,}50 * 10^{12}) D_i s \phi}$$

$$\Delta P_r = 1{,}334 * 10^{-13} \alpha_r G^2 / s \quad (Velocity\ Head)$$

Nozzles ΔP

$$\Delta P_n = 2 * 10^{-13} N_s {G_n}^2 / s \quad (Turbulent\ flow)$$

$$\Delta P_n = 4 * 10^{-13} N_s {G_n}^2 / s \quad \text{(Laminar flow)}$$

α_r		
Flow Regime	Regular Tubes	U-tubes
Turbulent	2Np-1.5	1.6Np-1.5
Laminar	3.25Np-1.5	2.38Np-1.5

Shell side $\Delta \boldsymbol{P_{shell}} = \Delta P_f + \Delta P_n$

$$\Delta P_f = \frac{f G^2 d_s (n_b + 1)}{(7{,}50 * 10^{12}) d_e s \phi}$$

Nozzles ΔP

$$\Delta P_n = 2 * 10^{-13} N_s {G_n}^2 / s \quad (Turbulent\ flow)$$

$$\Delta P_n = 4 * 10^{-13} N_s {G_n}^2 / s \quad \text{(Laminar flow)}$$

For a suitable Heat Exchanger evaluation

$$\Delta \boldsymbol{P_{tubes,allowed}} > \Delta \boldsymbol{P_{tubes}}$$

$$\Delta \boldsymbol{P_{shell,allowed}} > \Delta \boldsymbol{P_{shell}}$$

HEAT EXCHANGER:CONSTRAINTS

Factors affectting pressure drop

Tube Side

1. Tube Length (L)
2. Number of tube passes n_p

Shell Side

1. Baffle spacing (B) . Increasing B increases the flow area across the tubes bundle which lowers the $\Delta \boldsymbol{P}_{\boldsymbol{shell}}$
2. Tube pitch (P_T). It is not common used because increasing the tubes pitch increases the heat exchanger area and therefore its cost.

Theses factors are important when designing both in hand made and computer aided softwares. In case the inital geometry chosen does not meet the pressure drop requirements, then it is neccesary to change the factors that affect potentially the pressure drop across the heat exchanger in order to reach a suitable equipement.

HEAT EXCHANGER:INITIAL GEOMETRY

But…What initial geometry must I choose? [4]

1

FLUID PLACEMENT(TUBE SIDE)
Cooling water
The more fouling
The less viscous
The higher pressure
The hotter fluid
The smaller volumetric flowrate

2

TUBING SELECTION				
Service	Size(in)	BWG	L(ft)	Pitch(in)
Water	3/4	16	16-20	1
Hydrocarbon(Low fouling)	3/4	14	16-21	1
Hydrocarbon(High fouling)	1	14	16-22	1.25
Pitch can be triangular or square. For high fouling fluids it is recommended to use the square pitch				

4

BAFFLES	
Type	Single segmental (widely used)
Spacing	0.2ds-1ds
Cut	20-35%(20% recommended for Delaware method)
Thickness(in)	(1/16)-(3/4)

3

SHELL	
Type	Applications
E	Standard
F	Two shell pass flow (truely counter flow)
G,H,K,X	Reboilers,Condensers,Coolers
J,K	When low ΔP(shell) is required

HEAD		
Property	**Bonnet**	**Channel**
Cost	Cheaper	More Expensive
Prone to leakage	Less probability	High probability
Access	Disconnect process piping and remove from shell	Unbolting and removing channel cover(easier)
	Fixed tubesheet	**Floating tubesheet**
Cost	Cheaper	More Expensive
Prone to leakage	Less probability	High probability
Cleaning	Cannot be removed for cleaning	Can be removed for cleaning
A floating head and U-tubes exchangers can be used when mechanical cleaning is needed		

HEAT EXCHANGER:INITIAL GEOMETRY

But…What initial geometry must I choose? [4]

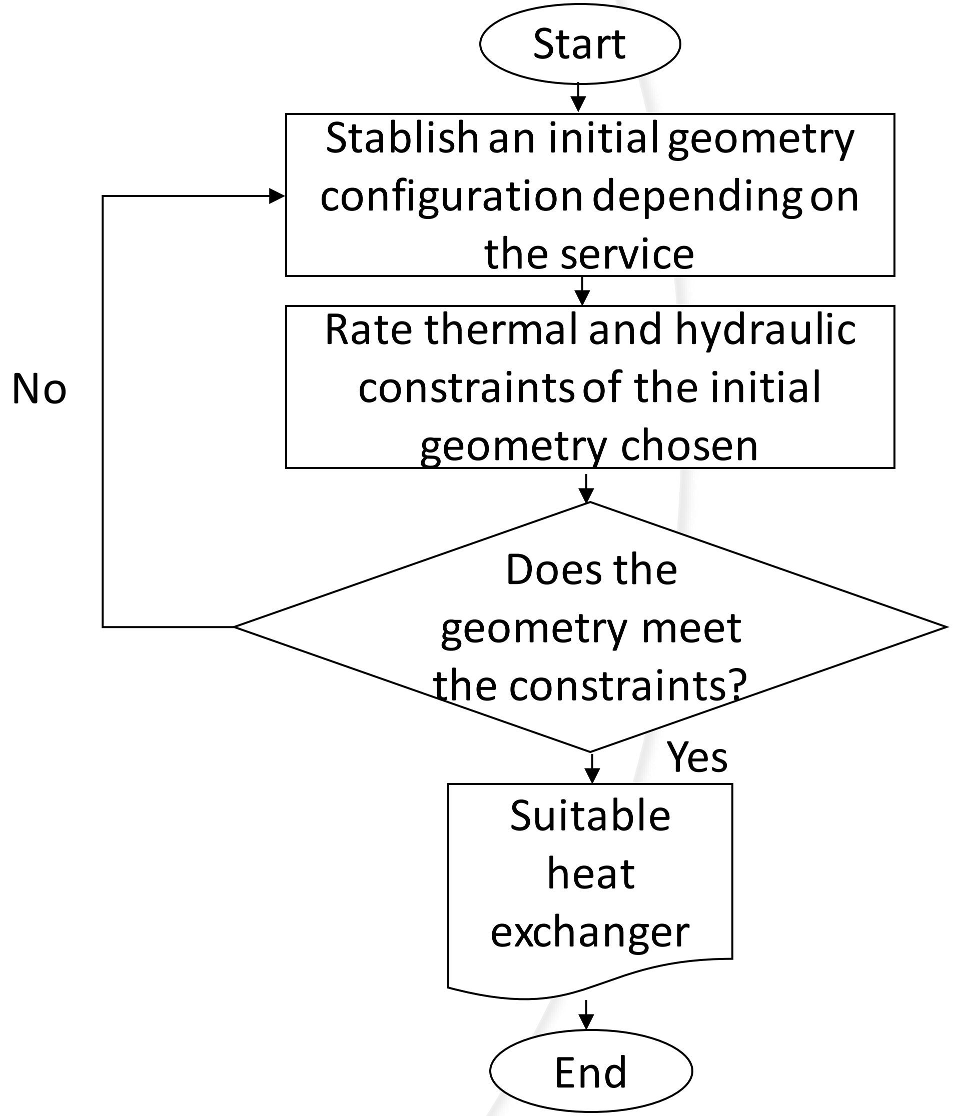
5

NOZZLES	
Shell size(in)	**Nominal Diameter(in)**
4-10	2
12-17.25	3
19.25-21.25	4
23-29	6
31-37	8
39-42	10

6

SEALING STRIPS
One pair/10 tubes rows

Shell and Tube Design Flow diagram[4]



HEAT EXCHANGER:EXAMPLE

A kerosene stream with a Flow rate of 45000 lb/h is to be cooled from 390°F to 250°F by heat Exchange with 150000 lb/h of crude oil at 100°F. A máximum pressure drop of 15 psi has been specified fro each stream. Prior experience with this particular oil indicates that it exhibits significant fouling tendencias, and a fouling factor of 0,003 $^{h\,ft^{2}{}^{\circ}F}/_{BTU}$ is recommended. Design a Shell and Tube Heat Exchanger for this application.

PHYSICAL PROPERTIES		
Fluid property	**Kerosene**	**Crude Oil**
Cp(BTU/lbm°F)	0,59	0,49
k(BTU/h ft °F)	0,079	0,077
μ(lbm/ft h)	0,97	8,5
Specific gravity	0,785	0,85
Pr	7,24	55,36

Initial Geometry		
Tube fluid	Crude Oil	
TEMA Configuration	AES	
Tubing Selection	Tube Size(in)	1
	Tube BWG	14
	Tube Long(ft)	20
	Tube Layout	Square
	Tube Pitch(in)	1,25
Baffles	Cut	20%
	Spacing(B/ds)	0,3
Sealing Strips	Pair/10 tubes rows	1
Material	Shell and tube side	Carbon steel

Example taken from [4]

HEAT EXCHANGER:HTRI

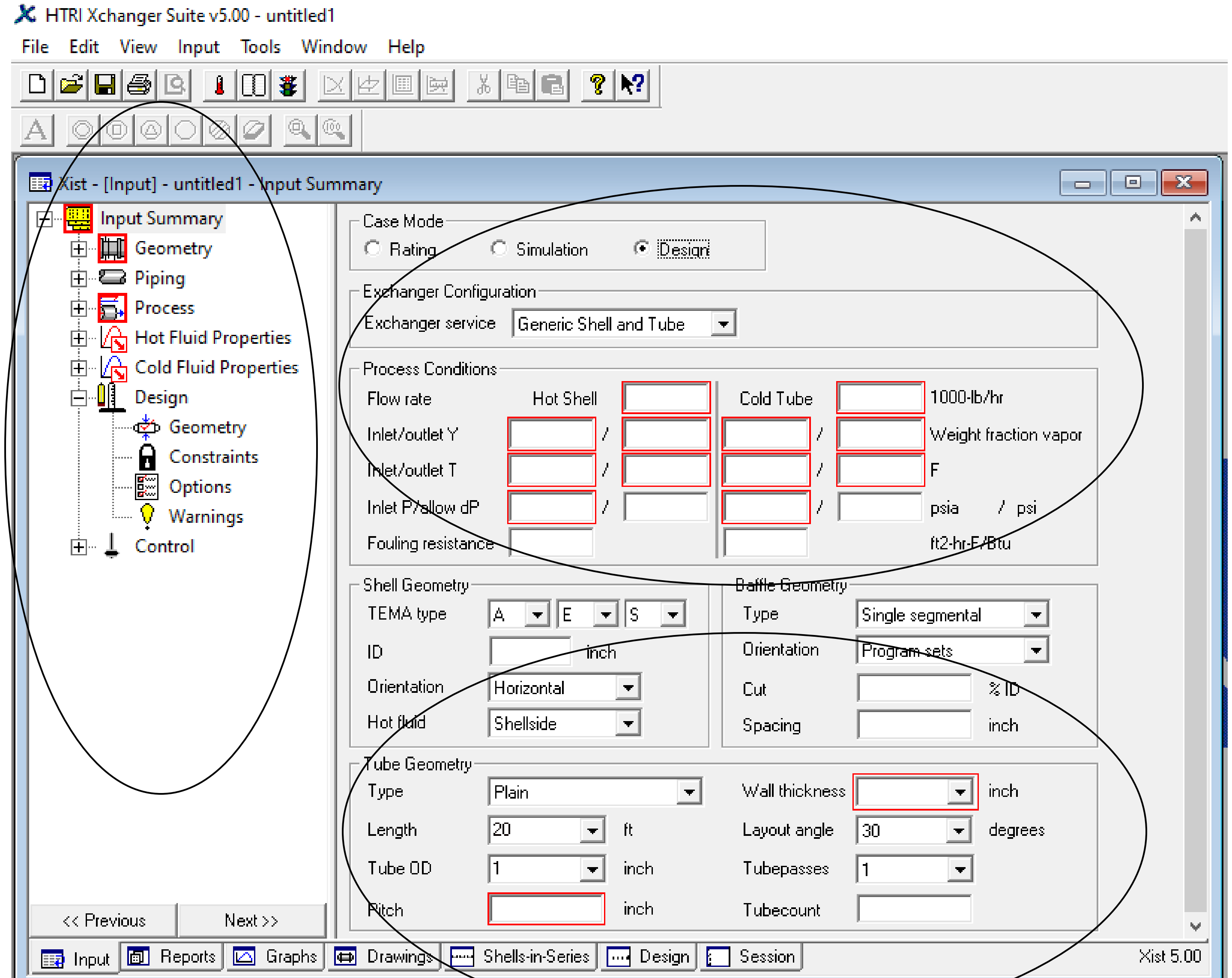
- ✓ As many of you might know, HTRI is a software of engineering used for the simulation,rating and design of heat exchangers.
- ✓ When entering to the interface the program offers a wide variety of heat exchangers, in which Shell and tube heat exhanger is found.
- ✓ It is neccesary to know basics of heat exhanger design previously, because when running your case probably your design will not reach a solution due to thermall and/or hydraulic contraints did not meet the initially conditions given.
- ✓ Most of the cases for not reaching a solution is because the pressure drop calculated is greater than the pressure drop allowed. In those cases, it is necessary to change the factors that affect potentially the pressure drop across tubes and/or shell side until finding a design which can meet the over-design allowed by your client.
- ✓ After running your case, if any change needed, the program suggests to change some parameters. These suggestions can be as: fatal messages or warning messages. Both are important.
- ✓ A feature HTRI offers is that you can design using other constraints such as: Tube long, tube and Shell passes; bafle spacing, tube diameter, etc. However, it is important to know that when tightening too much your case, the design could not be reached easier, because the program iterations did not reach a solution according to these constraints given.
- ✓ HTRI does not have a large component list as other programs have. Neverthless, properties can also be saved and provided by the user.

HEAT EXCHANGER:HTRI INTERFACE

After

1. knowing the input geometry (guessed)
2. Filling all boxes required.
3. Troubleshooting the initial run warnings

Solution is reached



HEAT EXCHANGER:HTRI RESULTS

HTRI — **HEAT EXCHANGER SPECIFICATION SHEET** — Page 1 — **US Units**

		Job No.	
Customer		Reference No.	
Address		Proposal No.	
Plant Location		Date 30/07/2020	Rev
Service of Unit		Item No.	
Size 15,2500 x 240,000 inch	Type AES Horz.	Connected In 1 Parallel	1 Series
Surf/Unit (Gross/Eff) 418,88 / 413,56 ft2	Shell/Unit 1	Surf/Shell (Gross/Eff) 418,88 / 413,56 ft2	

PERFORMANCE OF ONE UNIT

Fluid Allocation		Shell Side		Tube Side	
Fluid Name		Kerosone		Crude Oil	
Fluid Quantity, Total	lb/hr	45000,0		150000	
Vapor (In/Out)					
Liquid		45000,0	45000,0	150000	150000
Steam					
Water					
Noncondensables					
Temperature (In/Out)	F	390,00	250,00	100,00	150,57
Specific Gravity		0,7800	0,7800	0,8446	0,8446
Viscosity	cP	0,4010	0,4010	3,5965	3,5965
Molecular Weight, Vapor					
Molecular Weight, Noncondensables					
Specific Heat	Btu/lb-F	0,5900	0,5900	0,4900	0,4900
Thermal Conductivity	Btu/hr-ft-F	0,0790	0,0790	0,0770	0,0770
Latent Heat	Btu/lb				
Inlet Pressure	psia				
Velocity	ft/sec	1,70		5,21	
Pressure Drop, Allow/Calc	psi	15,000	7,188	15,000	5,443
Fouling Resistance (min)	ft2-hr-F/Btu	0,00300		0,00300	

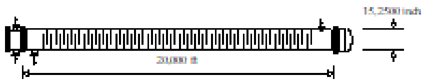
Heat Exchanged Btu/hr 3716999 — MTD (Corrected) 182,2 F

Transfer Rate, Service 49,33 Btu/ft2-hr-F — Clean 92,86 Btu/ft2-hr-F — Actual 57,58 Btu/ft2-hr-F

CONSTRUCTION OF ONE SHELL

		Shell Side	Tube Side
Design/Test Pressure	psig	150,000 /	150,000 /
Design Temperature	F		
No Passes per Shell		1	2
Corrosion Allowance	inch		
Connections Size & Rating	In inch	1 @ 2,4690	1 @ 4,0260
	Out inch	1 @ 4,0260	1 @ 3,0680
	Intermediate	@	@

Sketch (Bundle/Nozzle Orientation)



Tube No. 80	OD 1,0000 inch	Thk(Avg) 0,0830 inch	Length 20,000 ft — Pitch 1,2500 inch — Layout 60
Tube Type	Plain		Material CARBON STEEL
Shell	ID 15,2500 inch	OD inch	Shell Cover
Channel or Bonnet			Channel Cover
Tubesheet-Stationary			Tubesheet-Floating
Floating Head Cover			Impingement Plate — Circular plate
Baffles-Cross	Type SINGLE-SEG.	%Cut (Diam) 20,0	Spacing(c/c) 3,0777 — Inlet 16,7036 inch
Baffles-Long		Seal Type	
Supports-Tube		U-Bend	Type
Bypass Seal Arrangement		Tube-Tubesheet Joint	
Expansion Joint		Type	
Rho-V2-Inlet Nozzle	2904,14 lb/ft-sec2	Bundle Entrance 34,62	Bundle Exit 64,94 lb/ft-sec2
Gaskets-Shell Side		Tube Side	
-Floating Head			
Code Requirements			TEMA Class
Weight/Shell 4425,15		Filled with Water 5987,20	Bundle 1810,13 lb
Remarks:			

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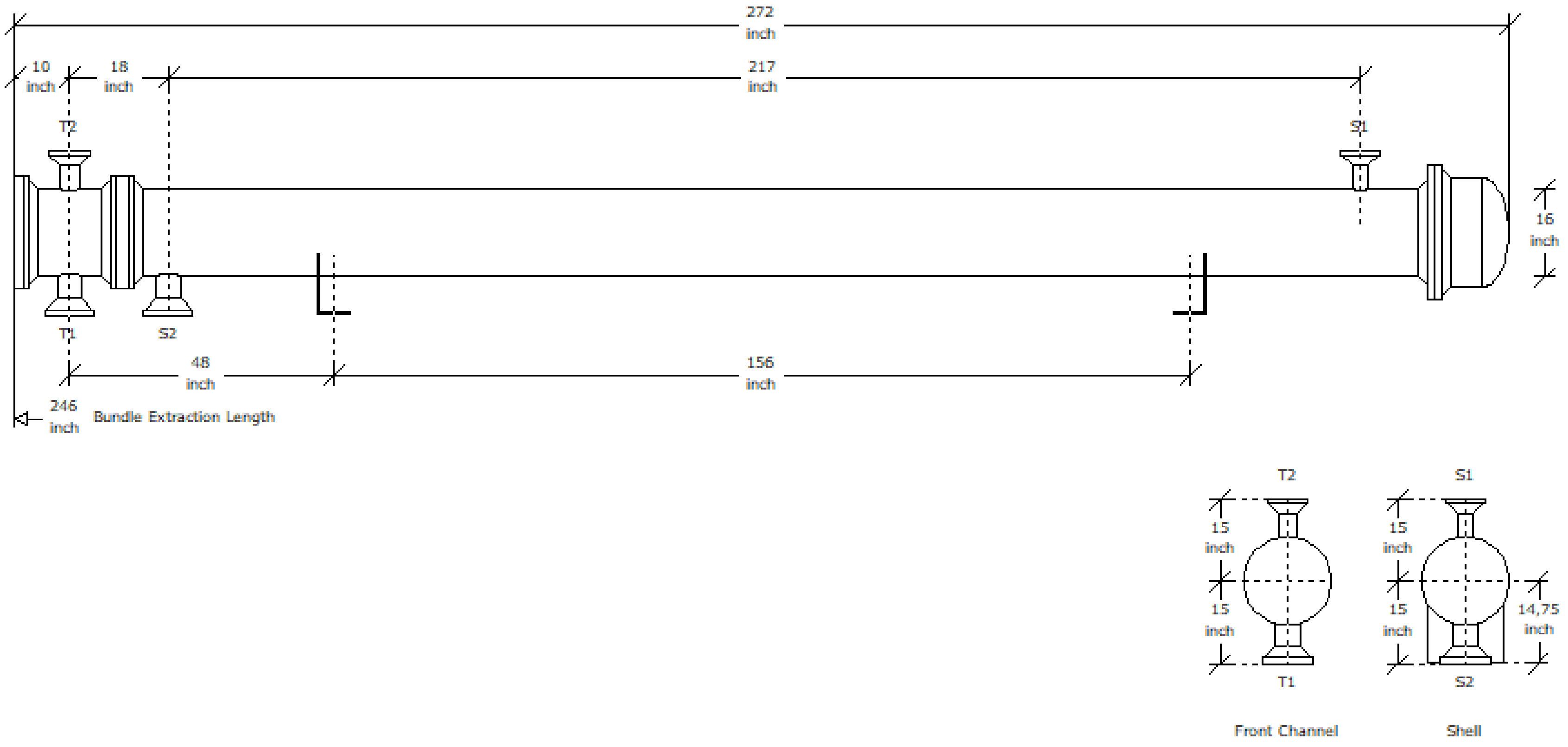
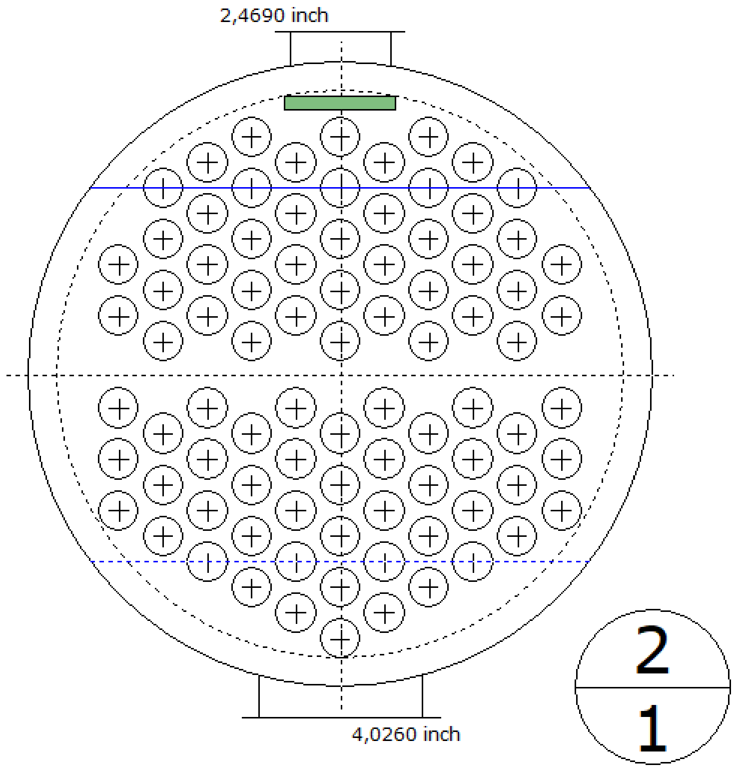
Case	Over Design %	Total Area (ft2)	Duty (MM Btu/hr)	EMTD (F)	U (Btu/ft2-hr-F)	Shell h (Btu/ft2-hr-F)	Tube h (Btu/ft2-hr-F)	Shell Velocity (ft/sec)	Tube Velocity (ft/sec)	Shell DP (psi)	Tube DP (psi)	Shells In Series	Shells In Parallel	Shell ID (inch)	Baffle Spacing (inch)	Tube Passes	Tube Length (ft)
Design ...	17,21	397,935	3,7170	181,2	61,21	304,75	194,38	1,67	5,48	7,117	5,824	1	1	15,2500	3,0777	2	20,000

HEAT EXCHANGER:HTRI RESULTS



	Nozzles	NPS, in	Rating	Design	Shell	Tube	Weight	lb	Company	Juan		
S1	Inlet	2,5	150	Pres (psig)	150	150	Bundle	2300	Customer	None	Ref	00
S2	Outlet	4	150	Temp (F)			Dry	5200	Item			
T1	Inlet	4	150	Passes	1	2	Wet	7000	Service	Kerosene-Oil Shell & Tube Heat Exchanger		
T2	Outlet	3	150	Thick (inch)	0,375	0,083			TEMA	AES		Setting Plan
									Date	30/07/2020	By	
									Diagram		Rev	0

HEAT EXCHANGER:HTRI RESULTS



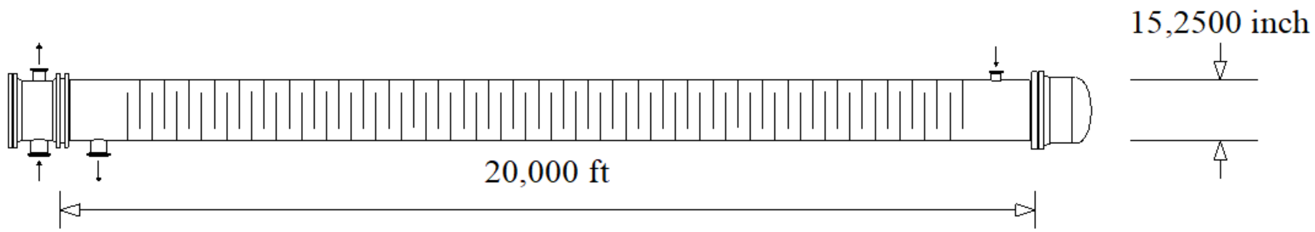
Item number		
TEMA type	AES	
Shell diameter	15,2500	inch
Outer tube limit	13,8992	inch
Height under inlet nozzle	0,8246	inch
Height under outlet nozzle	0,6754	inch
Tube diameter	1,0000	inch
Tube pitch	1,2500	inch
Tube layout angle	60	
Number of tubes (specified)	80	
Number of tubes (calculated)	80	
Number of tie rods	6	
Number of seal strip pairs	2	
Number of passes	2	
Perpendicular passlane width	0,6250	inch
Baffle cut % diameter	20	

TUBEPASS DETAILS

Pass	Rows	Tubes	Plugged
1	10	43	0
2	9	43	0

SYMBOL LEGEND

- ○ Tube
- Plugged tube
- Tie rod
- Impingement rod
- Dummy tube
- ● Seal rod
- ▭ Seal strip/Skid bar



HEAT EXCHANGER:ASPEN EDR

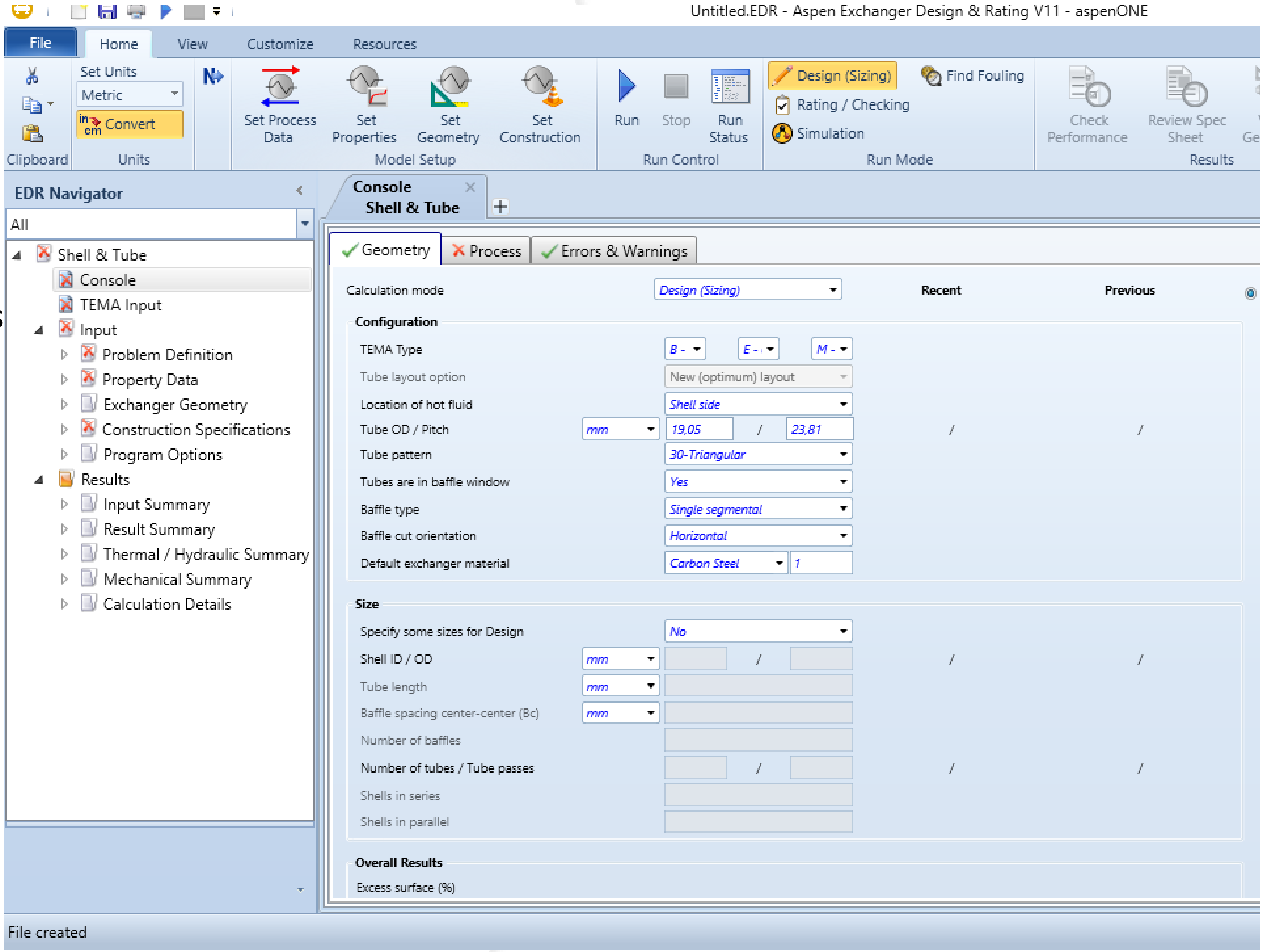
- ✓ As HTRI, Aspen EDR can also be used for the simulation,rating and design of heat exchangers.
- ✓ When entering to the interface the program offers a wide variety of heat exchangers, in which Shell and tube heat exhanger is found.
- ✓ It is neccesary to know basics of heat exhanger design previously, because when running your case probably your design will not reach a solution due to thermall and/or hydraulic contraints did not meet the initially conditions given.
- ✓ Most of the cases for not reaching a solution is because the pressure drop calculated is greater than the pressure drop allowed. In those cases, it is necessary to change the factors that affect potentially the pressure drop across tubes and/or shell side until finding a design which can meet the over-design allowed by your client.
- ✓ A feature Aspen EDR offers is that you can design using other constraints such as: Tube long, tube and Shell passes; bafle spacing, tube diameter, etc. However, it is important to know that when tightening too much your case, the design could not be reached easier, because the program iterations did not reach a solution according to these constraints given.
- ✓ As a suite of Aspen, Aspen EDR is interconnected with hysys and you can pass from hysys to EDR(viceversa) without problems and without filling all physical properties data. (In this case, it was needed becase “Crude Oil” doesn’t exist in the list. However,if it was needed to start from hysys, it could also be added as an hypothetical component)
- ✓ Aspen EDR takes less time in doing the iterations.

HEAT EXCHANGER:EDR INTERFACE

After

1. knowing the input geometry (guessed)
2. Filling all boxes required.
3. Troubleshooting the initial run warnings

Solution is reached



HEAT EXCHANGER:EDR RESULTS

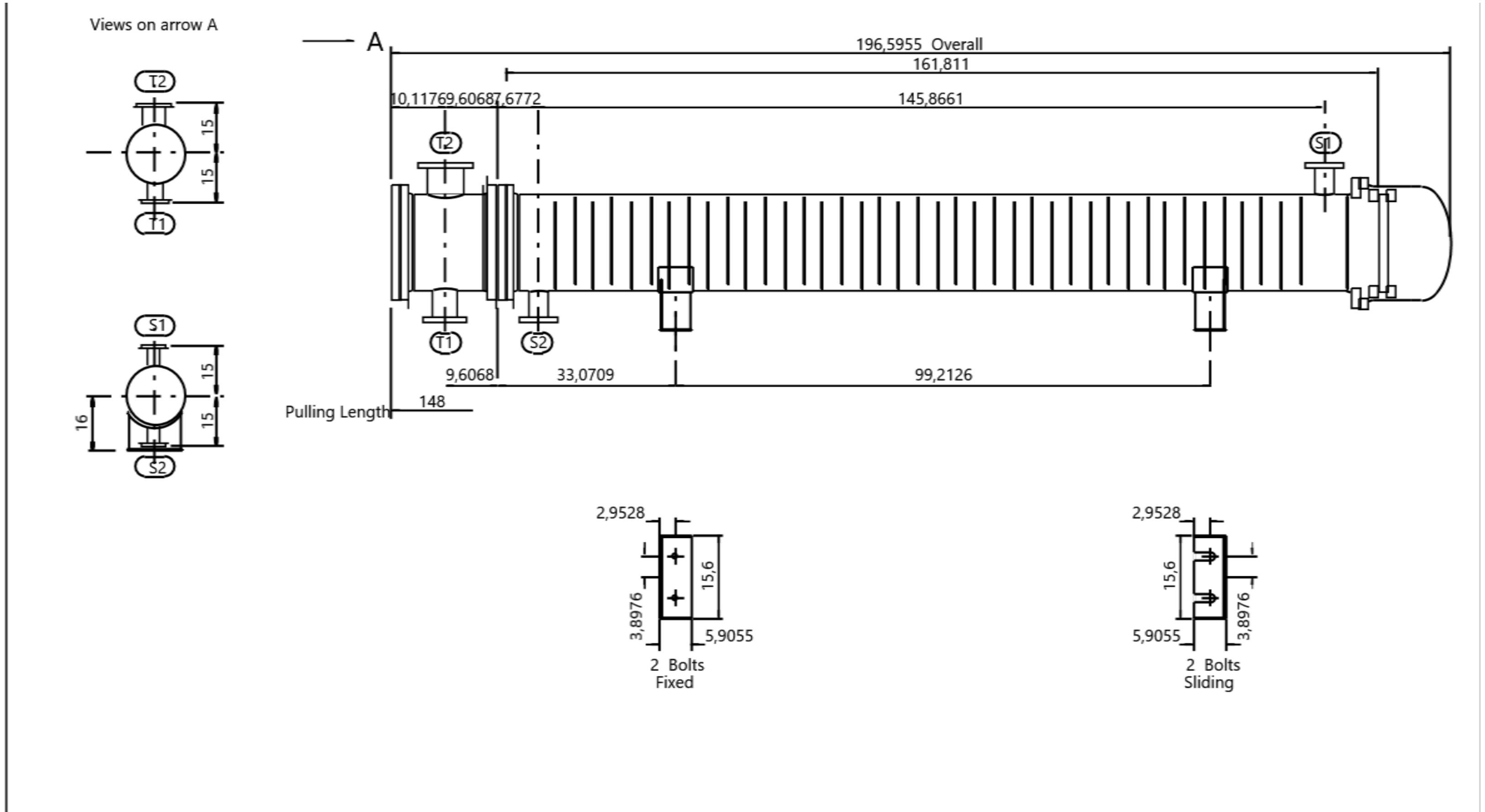
TEMA Sheet

Heat Exchanger Specification Sheet

1	Company:					
2	Location:					
3	Service of Unit: Our Reference:					
4	Item No.: Your Reference:					
5	Date: Rev No.: Job No.:					
6	Size : 438 - 4200 mm	Type: AES Horizontal		Connected in: 1 parallel 1 series		
7	Surf/unit(eff.) 320,7 ft²	Shells/unit 1		Surf/shell(eff.) 29,8 m²		
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		Shell Side		Tube Side	
10	Fluid name					
11	Fluid quantity, Total	lb/h	45000		150000	
12	Vapor (In/Out)	kg/h	0	0	0	0
13	Liquid	lb/h	45000	45000	150000	150000
14	Noncondensable	kg/h	0	0	0	0
15						
16	Temperature (In/Out)	°F	390	250	100	150,57
17	Bubble / Dew point	°F	/	/	/	/
18	Density Vapor/Liquid	lb/ft³	/ 48,67	/ 48,67	/ 52,7	/ 52,7
19	Viscosity	cp	/ 0,401	/ 0,401	/ 3,5964	/ 3,5964
20	Molecular wt, Vap					
21	Molecular wt, NC					
22	Specific heat	BTU/(lb-F)	/ 0,59	/ 0,59	/ 0,49	/ 0,49
23	Thermal conductivity	BTU/(ft-h-F)	/ 0,079	/ 0,079	/ 0,077	/ 0,077
24	Latent heat	kcal/kg				
25	Pressure (abs)	psi	50	47,39	50	36,62
26	Velocity (Mean/Max)	m/s	0,24 / 0,52		2,71 / 2,89	
27	Pressure drop, allow./calc.	psi	15	2,61	15	13,38
28	Fouling resistance (min)	ft²-h-F/BTU	0,003		0,003 0,0036 Ao based	
29	Heat exchanged 3717064	BTU/h		MTD (corrected) 102,71 °C		
30	Transfer rate, Service 62,69		Dirty 63,71	Clean 109,9 BTU/(h-ft²-F)		

31	**CONSTRUCTION OF ONE SHELL**				**Sketch**
32			Shell Side	Tube Side	
33	Design/Vacuum/test pressure	psi	58,02 / /	58,02 / /	
34	Design temperature / MDMT	°F	455 /	455 /	
35	Number passes per shell		1	4	
36	Corrosion allowance	mm	3,18	3,18	
37	Connections	In mm	1 76,2 / -	1 101,6 / -	
38	Size/Rating	Out	1 76,2 / -	1 152,4 / -	
39	Nominal	Intermediate	/ -	/ -	
40	Tube #: 94 OD: 1 Tks. Average 0,083 in Length: 4200 mm Pitch: 1,25 in Tube pattern: 90				
41	Tube type: Plain Insert: None Fin#: #/m Material: Carbon Steel				
42	Shell Carbon Steel ID 17,25 OD 18 in			Shell cover Carbon Steel	
43	Channel or bonnet Carbon Steel			Channel cover Carbon Steel	
44	Tubesheet-stationary Carbon Steel -			Tubesheet-floating Carbon Steel	
45	Floating head cover Carbon Steel			Impingement protection None	
46	Baffle-cross Carbon Steel Type Single segmental Cut(%d) 7,34 H Spacing: c/c 3,5433 in				
47	Baffle-long - Seal Type			Inlet 9,1029 in	
48	Supports-tube U-bend 0 Type				
49	Bypass seal Tube-tubesheet joint Expanded only (2 grooves)(App.A 'i')				
50	Expansion joint - Type None				
51	RhoV2-Inlet nozzle 1813 Bundle entrance 56 Bundle exit 56 kg/(m-s²)				
52	Gaskets - Shell side Flat Metal Jacket Fibe Tube side Flat Metal Jacket Fibe				
53	Floating head Flat Metal Jacket Fibe				
54	Code requirements ASME Code Sec VIII Div 1 TEMA class R - refinery service				
55	Weight/Shell 1605,8 Filled with water 2300,8 Bundle 737,5 kg				
56	Remarks				
57					
58					

HEAT EXCHANGER:EDR RESULTS



Nozzle Data				
Ref	OD	Wall	Standard	Notes
S1	3,5"	0,216"	Slip on	
S2	3,5"	0,216"	Slip on	
T1	4,5"	0,237"	Slip on	
T2	6,625"	0,28"	Slip on	

Design Data	Units	Shell	Channel
Design Pressure	psi	58,01	58,01
Design Temperature	°F	455	455
Full Vacuum		0	0
Corrosion Allowance	in	0,125	0,125
Test Pressure	psi		
Number of Passes		1	4
Radiography		0	0
PWHT		0	0
Internal Volume	ft^3	15,7801	8,0893

Weight Summary		
Empty	Flooded	Bundle
3540 lb	5072 lb	1626 lb

Notes:

Scale:

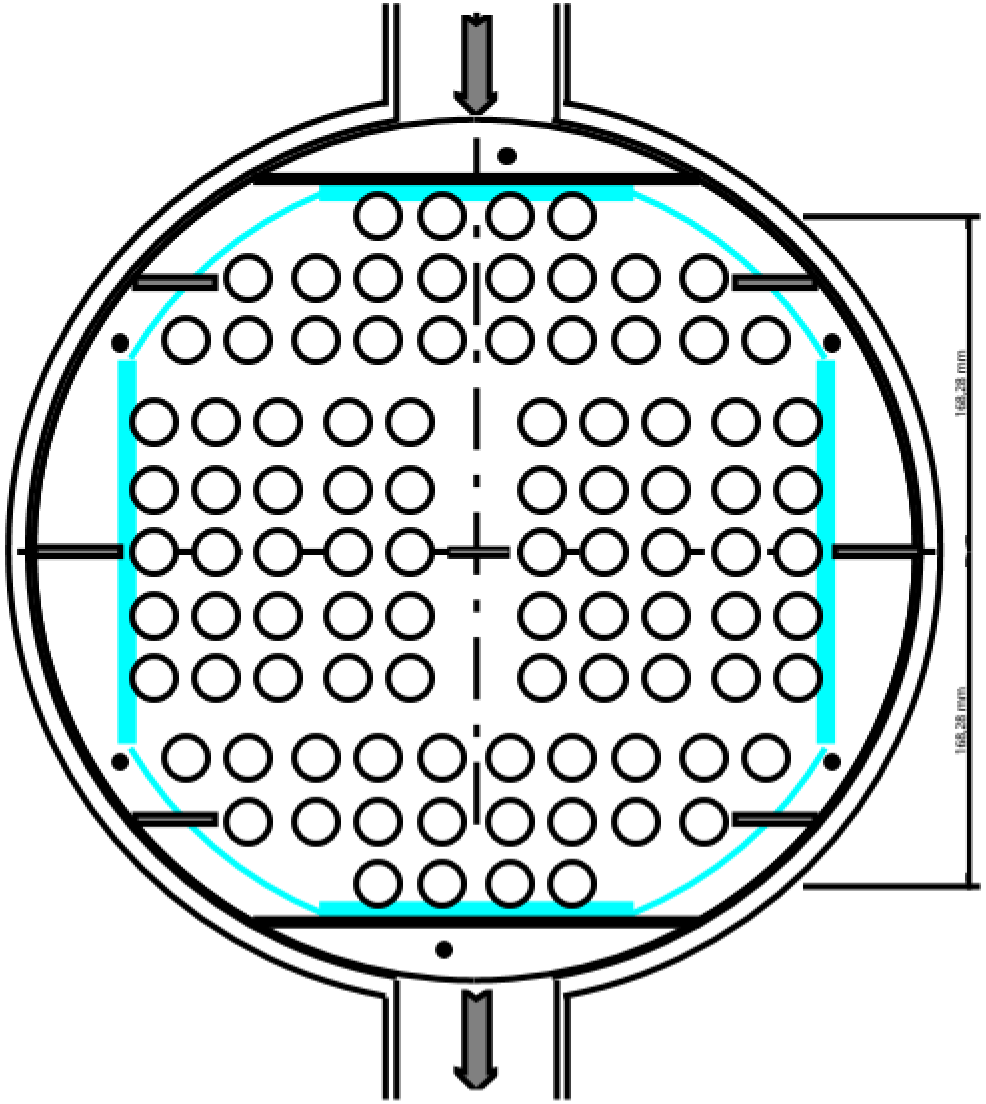
Rev:	Date:	Description	Dwg.	Chk.	Appd.

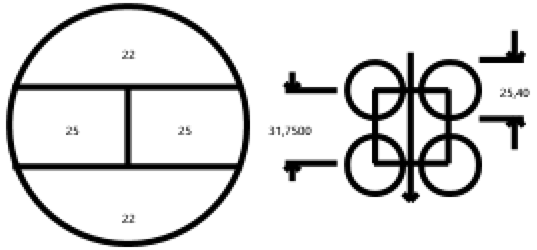
Company:
Location:
Service of Unit: Our Reference:
Item No.: Your Reference:
Date: Rev No.: Job No.:

Company Name
City, State

0	Setting Plan	
TEMA Type: AES		
Size: 17 - 165	Dwg No.:	Rev:
TEMA Class: 0	serth example 1	

HEAT EXCHANGER:EDR RESULTS





Shell inside diameter	mm	438,15
Front head inside diameter	mm	438,15
Outer tube limit	mm	393,7
Tube number (calcs.)		94
Tube number (layout)		94
Tube length	mm	4200
Tube O.D.	mm	25,4
Tube pitch	mm	31,75
Tube pattern		90
Tube passes		4
Tie rod number		6
Tie rod diameter	mm	9,55
Sealing strips (pairs)		3
Baffle type		Single segmental
Centre to outer baffle cut	mm	186,9281
Centre to inner baffle cut		
Impingement protection		None
Shell Side Inlet Nozzle Inside Diameter	mm	77,9272
Shell Side Outlet Nozzle Inside Diameter	mm	77,9272

Notes:

Scale:

Rev.	Date:	Description	Dwn.	Chk.	Appd.

Company:
Location:
Service of Unit: Our Reference:
Item No.: Your Reference:
Date: Rev No.: Job No.:

Company Name
City, State

TEMA Type: AJS
Size: 438 - 4200

Tube Layout

Dwg No.: Untitled 2
Rev:

HEAT EXCHANGER:ANALYSIS

SHELL AND TUBE HEAT EXCHANGER DESIGN COMPARISION				
RESULTS		**SERTH (HAND MADE)**	**HTRI**	**EDR**
Shell	**Fluid**	Kerosene	Kerosene	Kerosene
	Type	AES	AES	AES
	ID(in)	19,25	15,25	17,25
Tubes	**Fluid**	Crude Oil	Crude Oil	Crude Oil
	Number of tubes	124	80	94
	Size OD(in)	1	1	1
	Tube BWG	14	-	-
	Tube Length(ft)	14	20	13,77
	Tube Layout	Square	Triangular	Square
	Tube Pitch(in)	1,25	1,25	1,25
	Tube Passes	4	2	4
Heat transfer area	**ft2**	454	397,935	320,7
Baffles	**Cut(%)**	20	20	7,34
	Spacing(in)	3,85	3,0777	3,54
Sealing Strips	**Pair/10 tubes rows**	1	2	3
Nozzles	**Tube side**	4 in Sch 40	T1(4 in, CL 150) T2(3 in, CL150)	T1(4,5in) T2(6,62 in)
	Shell side	3 in Sch 40	S1(2.5 in, CL 150) S2(4 in, CL150)	S1(3,5in) S2(3,5 in)
Material	**Shell and tube side**	Carbon steel	Carbon steel	Carbon steel

HEAT EXCHANGER:ANALYSIS

Which design is the best one?

- ✓ First at all, all methods used reached a solution according to the constraints initially stablished. However, it must be highlighted that each method can be analized according to the needs of the client.
- ✓ There are cases in which tube length must be carefully designed due to the fact sometimes the space available at plant is not enough for long tube length.
- ✓ If the cost of the equipment was a concern then the lowest area should be chosen (EDR)
- ✓ Despite Aspen EDR reached a solution. Between all methods, it would not be preferable to choose EDR because its tube side pressure drop would be a problem in the future because at clean condition (new) is just at 1,62 psi from the pressure drop allowance. An eventual obstruction or increasing of fouling would put in danger the equipment.
- ✓ As said in other presentations, hand made gave good results. However, this method tends to fall into calculation mistakes.

One of the main advantage of EDR it is that can be used integrated inside process simuation developed in HYSYS to design/evaluate rigorous the performance of heat exchangers. However, thermal departments of engineering companies always are truly about HTRI results

As recommendation, EDR in design mode can be used to provide a good initial estimation point to initiate the heat transfer calculation with HTRI. [5]

TERMINOLOGY

Parameter	Definition
Q	Rate of heat transfer
m	Mass flow
Cp	Heat capacity
F	LMTD correction factor
ΔTm	Logarithmic mean temperature difference
U_{Req}	Required overall heat transfer coefficient
U_{clean}	Clean overall heat transfer coefficient
U_D	Design overall heat transfer coefficient
D_o	External pipe diameter
D_i	Internal pipe diameter
k	Pipe thermal conductivity
R_{Di}	Fouling factor inner fluid
R_{Do}	Fouling factor outer fluid
h_i	Heat transfer coeffficient for inner fluid
h_o	Heat transfer coeffficient for outer fluid
ΔP_{Shell}	Shell pressure drop
ΔP_{tubes}	Tubes pressure drop
ΔP_f	Pressure drop due to fluid friction
ΔP_r	Pressure drop due to return bends
ΔP_n	Pressure loss in nozzles
f	Darcy friction factor
n_p	Tubes number of passes
L	Tube lenght
G	Mass flux
α_r	Number of velocity head allocated for minor losses in tube side
P_T	Tubes pitch
d_s	Shell ID
n_b	Number of baffles
d_e	Equivalent diameter
s	Fluid specific gravity
ϕ	Viscosity correction factor

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[3] Tubular Exchanger Manufacturers Association, Inc, Standars of the Tubular Exchanger Manufacturers Association 9th Edition, Tarrytown,NY, 2007.

[4] R. W. Serth, Process Heat Transfer: Principles and Applications, Texas USA: Elsevier, 2007.

[5] M. Gutierrez, *EDR or HTRI,* USA: LinkedIn, 2017.

¡Thanks for watching!

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