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Computer Aided Design of 1-2 Vertical Condenser

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ABSTRACT
Heat exchangers are used to transfer thermal energy between two or more media and are widely applicable to chemical industries, food industries, power engineering and so on. The shell and tube exchanger is the most common heat exchanger type hence the study of its design and analysis is of great importance for conserving energy in heat exchange process. The aim of this research is to design a 1-2 vertical condenser with the aid of a computer. This work is motivated due to its application in offshore location and also is suited for higher pressure applications. The objectives include using mathematical models, Matlab ‘m file design program and plat form independent java program in designing a 1-2 vertical condenser which is useful to the engineers. The work was carried out by calculating the heat transfer surface, condensate loading on vertical tube, mass velocity of shell and tube, Reynold number of shell and tube, flow area of shell, pressure drops in shell and tube, baffle space, mass velocity of water flowing in the tube, tube length, temperature of tube wall and film, and the dirt factor for five different mass flow rates (2.52kg/s, 5.04kg/s) 7.56kg/s 10.08kg/s and 12.60kg/s) of pure n- propanol coming out from the top of a distilling column. The results obtained showed that for the first mass flow rate (2.52kg/s), the pressure drops in shell and tube were 17236.89N/M^2 and 4103.64N/M^2 respectively, with dirt factor or of 0.000986M^2 K/W. For the second mass flow rate, it was 17236.89N/M^2 and 21110.13N/M^2 respectively with dirt factor of 0.000854M^2 K/W. For the third mass flow rate, it was 17236.89N/M^2 and 54995.83N/M^{2}respectively with dirt factor of 0.000625M^{2}k/W. For the fourth mass flow rate, the pressure drop were 17236.N/M^{2}and 107006.635N/M^2 respectively, with dirt factor of 0.000493 M^{2}K/W. For the fifth mass flow rate, it was 17236.89N/M^2 and 185744.766N/M^2 respectively with dirt factor of 0.000366M^{2}K/W. The first three condensers are somewhat suitable from heat transfer standpoint because calculated pressure drops did not exceed allowable pressure drops and also calculated dirt factor exceed required dirt factor while the last two condensers are unsuitable from heat transfer standpoint because calculated pressure drops exceed allowable pressure drops and also calculated dirt factor did not exceed required dirt factor.

Keywords: thermal energy, mathematical models, heat transfer, condenser

INTRODUCTION

The modification of vapor to liquid is termed condensation. Whereas for a pure fluid compound at a given pressure, the modification from vapor to liquid happens at however one temperature that is the saturation or equilibrium temperature. Since vapor- liquid heat transfer changes sometimes occur at constant or nearly constant pressure in trade, the condensation of one compound usually happens isothermally. Condensation happens at terribly completely different rates of warmth transfer by either of the 2 distinct physical mechanisms, that are drop wise and film wise condensation. The compression film constant is influenced by the feel of the surface on that condensation happens and conjointly by whether or not the compression surface is mounted vertically or horizontally. In spite of those apparent complications condensation like flow lends itself to direct mathematical study (J. Willard chemist 1950).

In 1-2 vertical condensers, condensation within vertical tubes follows primarily an equivalent mechanism as condensation outside vertical tubes if the interference of the shell baffles is neglected. Since the condensation film has the flexibility to grow ceaselessly in its descent down the within or outside of the tubes, it's going to modification from contour to flow at some height between the highest and bottom. The native compression constant decreases ceaselessly from the highest down ward till at some purpose the film changes to flow. Once the transition to flow the constant will increase in accordance with the standard behavior of forced convection (colburn).

This research is just to work out the sizes of 1-2 vertical condenser which is able to condense completely different mass flow rates of pure propanol coming back from the highest of a distilling column operative at 103421.359N/M2 at that pressure it boils at 390.93K. Water at 302.59K are going to be used as cooling medium. a mud issue of zero.000529Sm2K/J is needed with allowable pressure drops of 13789.515N/M2 for the vapor and 68947.57299 N/M2 for the water and also the condenser uses zero.019M OD, 16BWG tube on zero.0238M triangular pitch. The general objective of this project work is to style a vertical n-propanol condenser mistreatment method conditions that embody that the speed at that heat passes from vaporThrough the liquid condensation film and into the cooling surface per unit space is given by Q/A = K (t1 – t) = W1 = h (t1 – t) wherever is (1.1). The heat of transformation of vaporization, W1 the pounds of condensation shaped per hour per square measure and y1 is that the thickness of the condensation film at the generalized purpose within the figure whose coordinates ar x1, y1 (Nusselt, 1916).

Statement of Problem

To quickly and accurately determine the sizes of 1-2 vertical condensers suitable for the condensation of five different mass flow rate (2.52kg/s, 5.04kg/s, 7.56kg/s, 10.08kg/s and 12.60kg/s) of substantially pure n-propanol (propyl alcohol) coming from the top of a distilling column operating at 10342.35n/m^2 at which pressure each boils at 390.93k. Water at 302.59k will be used as cooling medium. A dirt factor of 0.00053m^{2}k/w is required with allowable pressure drops of 13789.52 n/m^2 for the vapor and 68947.573n/m^2 for the water. Available for the service is 0.79m ID vertical 1-2 condenser with 0.019m OD, 16 BWG tube on 0.0238m triangular pitch.

Aim And Objectives Of The Study

The aim of this research work is to design a 1-2 vertical condenser with the aid of a computer.

Objectives include:

- To use mathematical models for the design of 1-2 vertical condenser which is useful to the engineers.
- To write a Matlab m-file design programs for the design of 1-2 vertical condenser which is also useful to the engineers.
- To developed a platform independent java programs for the design of 1-2 vertical condenser which is also useful to the engineers.

Different design variables like actual condensing film coefficient, heat transfer surface, condensate loading on vertical tubes, mass velocity of vertical condensers shell and tube, velocity of water flowing in tube, Reynold's number, dirt factor, tube length, baffle space, pressure drops in shell and tube, dirt factor, the temperature of tube wall and the temperature of film were obtained and plotted against the heat loads.

This is done by writing a Matlab M-file design programs and platform independent java programs for the various design variables of a 1-2 vertical condenser and comparing the results obtained with that of manual calculations. The results of the Matlab M-file design programs and platform independent java programs are the same with that obtained from manual calculations. Each of the values of the design variables are plotted against the heat loads and the nature of each graph is determined.

LITERATURE REVIEW

Drew et al (1935) represented that steam is that the solely pure vapor celebrated to condense in an exceedingly drop wise manner which special conditions square measure needed for its prevalence. They additionally discovered that these occurred thanks to the presence of dirt on the surface or presence of a stuff that adheres to the surface. The last word sort of associate equation for the compressing constant were obtained from dimensional analysis as an operate of the properties of the atmospheric phenomenon film, x, p, g, and L, t, and, the last being the latent heat of vaporization. NUSSELT (1916) in theory derived the relationships for the mechanism of film condensation, and also the results he obtained were bushed wonderful agreement with experiments. He created associate assumption that the warmth delivered by the vapor is heat solely which the drain of the atmospheric phenomenon film from the surface is by streamline flow solely, and also the heat is transferred through the film by conductivity.

Mc Adams (1940) found from the correlation of the information of many investigations that discovered compressing constants for steam on vertical tubes were seventy five per cent bigger than the theoretical coefficient calculate by a precise equation are evaluated at the film temperature t_f. The worth calculated from the on top of equation agrees but for a atmospheric phenomenon in stream line flow with the values calculated from another equation for standard flow.

By semi empirical means that Colburn (1934) has combined the impact of stream line flow within the higher portion of the tube thereupon of flow below the purpose wherever 4G1/F = 2100. This needed the choice of a heat transfer issue for forced convection specified h at the transition purpose were roughly a similar for each NUSSELT Condensation and flow. He then obtained the mean constant, for the complete height of the tube by coefficient the typical constant for the higher portion of the tube and h for forced convection in his lower portion of the tube.

Tinker (1951, 1958) revealed the primary careful stream- analysis methodology for predicting shell-side heat-transfer coefficients and pressure drop and also the ways afterward developed are supported his model. Tinker presentation is tough to follow and his methodology tough and tedious to use in manual calculations. It’s been simplified by Devore (1961, 1962) victimization normal tolerance for industrial exchangers and solely a restricted methodology in manual calculations. (Mueller, 1973) has additional simplified devore’s methodology and provides associate illustrative example. The engineering sciences knowledge unit has additionally revealed a technique for estimating shell-side pressure drop and warmth transfer constant, EDSU style guide 83038(1984). The tactic is predicated on a simplification of tinker's a technique. Bell (1960, 1963) developed a semi-analytical methodology supported work exhausted the cooperating analysis programme on shell and tube money changer at the University of Delaware. His methodology accounts for the key bypass and a escape streams and is appropriate for a manual calculation.

(Judge et al, 1997) have developed a heat exchange simulation for transient and steady state mixture and pure parts. Simulation is centered on air to refrigerant condenser and evaporator found in residential setup to quantify the impact of employing a zeotropic mixture R407C, with cross, parallel and counter flow device. consistent with this study once R407C is employed with a pure counter flow device the capability are often improved by four.4% for typical setup operational condition .experimental comparison between R22 and R407C has been done by (APREA et al, 2003), to guage the mechanical device performance in an exceedingly purpose engineered rig victimization semi-hermetic mechanical device. Constant of performance as a operate of outlet water temperature at the condenser shows that the over all energetic performance of R22 is within the vary 8%-14%. This study additionally shows that R407C has smaller actual meter potency thanks to smaller mass rate and lower is entropic mechanical device potency than that referring to R22.

(Chang et al, 2000) even have through an experiment investigated, the performance of a setup system victimization organic compound refrigerants. Propane, isobutane, butane, propylene, the only element hydrocarbons and propane/isobutane and propane/butane square measure the binary mixtures. The cooling and heating capacities of R290 square measure found slightly smaller than those of R22 with slightly higher constant of performance than of R22. The capability and COP (coefficient of performance) of R1270 square measure slightly bigger than R22 that is sweet indication for R1270 to be the attainable various for acquisition and warmth pump applications.

An extensive review of the recent progress in heat transfer sweetening victimization longitudinal vortex generation has been done by mathematician and Tai. As represented by Fiebig, (1998) once the attack angle is little, the generated vortices square measure in the main longitudinal, once the attack angle is 90°, crosswise vortices square measure generated. Mathematicians Tai and Fiebig, (1998) noted that Longitudinal vortices show less flow losses and higher heat transfer characteristics than crosswise vortices. In recent years, the third generation of the improved surface designed with longitudinal vortex generators (LVGS) is receiving additional attention.

The first report on longitudinal vortices in physical phenomenon management was conferred in 1960 by Schubauer and Spangenberg (1960). Johnson and Joubert (1969) early reported the impact of vortex generators on the warmth transfer performance in 1969. Since then, the utilization of LVGs in channel flow application has received wide attention.

Lee et al, (2003) numerically simulated heat transfer characteristics and turbulent structure in an exceedingly three-dimensional turbulent physical phenomenon with delta wing sort longitudinal vortices. The study found that longitudinal vortices ar capable of troubling the turbulent and thermal physical phenomenon, that caused turbulent intensity and augmentation of warmth transfer. aristocracy and Karl Gustav Jacob Jacobi (1816) through an experiment studied the warmth transfer sweetening performance of delta wings in an exceedingly flat plate flow by a naphthanlene sublimation technique. The results indicated that the common heat and mass transfer can be exaggerated by 50-60% at low. painter variety conditions compared with the consequences of associate degree external delta wing on the flow and warmth transfer characteristics in fan flows and uniform flows were through an experiment investigated and compared by subgenus Chen and Shu (2004) and Lau et al measured fast speed vector and temperature of the turbulent channel flow with rectangular winglets victimization quadruple start up probes.

Zhu et al (2000) computed the flow and warmth transfer in an exceedingly rectangular channel with rectangular winglets on one wall and rib-roughness parts on the opposite wall. The results showed that the combined result of rib-roughness and vortex generators might enhance the common Nusselt variety by nearly 450%. Sohankar, Simulated 3 dimensional unsteady flow and warmth transfer victimization. Direct numerical simulation (DNS) and huge Eddy simulation (LES) for an oblong channel with considering finite thickness of rectangular winglets at painter numbers of vary from two hundred to 2000. Deb et al (2004) numerically simulated heat transfer characteristics and flow ' structure in bedded and turbulent flows through an oblong channel containing inbuilt delta winglets.

Biswas et al (1994) applied numerical and experimental studies of flow structure and warmth transfer effects of longitudinal vortices behind a delta winglet placed in an exceedingly absolutely developed bedded channel flow. They found that Delta winglet turn out a main vortex, a corner vortex associate degreed an evoked vortex. Gang yong Lei et al (1990) have showed the consequences of baffle inclination angle on flow and warmth transfer of a device with whorled baffles, wherever the whorled baffles ar separated into inner and outer components on the radial direction of the shell. whereas each the inner and outer whorled baffle systematically, swimmingly and gently, and direct flow in an exceedingly whorled fashion thus on increase heat transfer rate and reduce pressure drop and impact vibrations, the outer whorled baffle becomes easier to manufacture because of its comparatively massive diameter of inner edge. Lutcha et al. have done experiments to the development of hollow heat exchangers with whorled baffles for investigation of the flow field patterns generated by varied helix angles that is predicted to say no pressure at shell aspect and increase heat transfer method considerably.

M. Sersa and A. Jimenez (2004) have conferred a compact formulation to relate the shell-side pressure drop with the money changer space and also the film constant supported the total Bell-Delaware technique. additionally to the derivation of the shell aspect compact expression, they need developed stream, that accounts for each straight pressure drops and come back losses. they need shown however the compact formulation may be used inside associate degree economical style rule. they need found a satisfactory performance of the projected algorithms over the whole pure mathematics vary of single section, shell and tube heat exchangers. Yusuf all Kara et al (2004), ready pc based mostly style model for preliminary style of shell and tube heat exchangers with single section fluid flow each on shell and tube aspect. The programme flow_both on shell and tube aspect. The programme determines the general dimensions of the shell, the tube bundle, and optimum heat transfer extent needed to satisfy the desired heat transfer duty by hard minimum or allowable shell aspect pressure drop. He over that current cold fluid in shell-side has some blessings on hot fluid as shell stream since the previous causes lower shell-side pressure drop and needs smaller heat transfer space than the latter and so it's higher to place the stream with lower mass rate of flow on the shell aspect thanks to the baffled area.

Wang et al (1999) applied associate degree experimental system for investigation on performance of shell and tube heat exchangers, and restricted experimental knowledge is obtained. The ANN (Artificial Neural Network) is applied to predict the temperature distinction and warmth transfer rate for warmth exchangers. Bp rule is employed to coach and take a look at the network. It's shown that the expected results are getting ready to experimental knowledge by ANN (Artificial Neural Network) approach. Comparison with correlation for prediction heat transfer rate shows ANN is superior to correlation indicating that ANN technique may be a appropriate tool to be used within the 'prediction of warmth transfer rates than empirical correlations. It's counseled that ANNS may be applied to simulate thermal systems, particularly for engineers to model the sophisticated heat exchangers in engineering applications.

Zahid Ayub (2000) conferred a replacement chart technique to calculate single-phase shell aspect heat transfer constant in an exceedingly typical TEMA tubuler money changer manufacturer association} vogue single segmental shell and tube device. A case study of rating water-to-water money changer is shown to point the result from this technique with core established procedures and software system accessible within the market. The results show that this new technique is reliable and similar to the foremost wide best-known HTRI software system.

Resat Selbas et al (2006) applied genetic algorithms (GA) for the best style of shell and tube device by varied the look variables: outer tube diameter, tube layout, variety of tube passes, outer shell diameter, baffle spacing and baffle cut. From this study, it had been over that the combinatorial algorithms like GA give important improvement within the best styles compared to the standard styles. GA application for deciding the world minimum neat money changer value is considerably quicker and has a plus over different ways in getting multiple resolution of same quality.

Master et al (2006) found that over half-hour heat exchangers square measure used of shell and tube sort. Shell and tube heat exchangers is bespoke by considering its operability, maintainability, flexibility and safety. This makes It terribly strong and serves major reason to be used wide in industries. He conclude that for economical heat transfer method, device ought to have depression drop, high shell facet mass speed, high heat transfer constant, and no or terribly Low fouling so on.

Lei et al (2008) dispensed a numerical investigation to review the impact of varied baffle inclination angles on fluid flow and warmth transfer of continuous spiral shell and lobe heat exchangers by exploitation periodic model. From the results computed, it absolutely was discovered that the best-integrated performance happens close to 45° angle. Performance of warmth money handler additionally depends on pressure drop. Outflow will cut back pressure drop and this per compartment average heat transfer constant.

Gaddis and Gnielisk (2010) projected a procedure to judge pressure drop and its comparison with experimental information.supported flow arrangement, Hari-Haran (2014) projected a simplified model for the study of thermal analysis of shell and tube sort heat exchangers of water and oil sort. The hardiness and medium weighted form of shell and tube heat exchangers build them similar temperament for top pressure operations. This paper shows the way to do the thermal analysis by exploitation theoretical formulae and for this, they need chosen a sensible drawback of counter flow shell and tube device of water and oil sort. By exploitation the information that return from theoretical formulae they designed a model of shell and tube device exploitation pro-E and done the thermal analysis by exploitation ANSYS (Analysis System) code and examination the result that obtained from ANSYS code and theoretical formulae.

Patel and Rao (2011) explores the employment of a untraditional optimisation technique; known as particle swarm optimisation (PSO), for style optimisation of shell and tube heat exchangers from economic read purpose. minimisation of total ANnual price is taken into account as an objective operate. 3 style variables like shell internal diameter, outer tube diameter and baffle spacing square measure thought of for optimisation. 2 tube layouts VIZ triangle and sq. also are thought of for optimisation. Four totally different case studies square measure given to demonstrate the Effectiveness and accuracy of projected rule. The results of optimisation exploitation PSO technique square measure compared with those obtained by exploitation genetic rule (GA).

El-Fawal et al (2011), given a computer virus for economical style of shell and tube device employing a pressure call order to reduce the price of the instrumentation. the planning procedure depends on exploitation the appropriate pressure drops so as to reduce the thermal extent for a particular service, involving separate call variables. Additionally the projected technique takes into account many geometric and operational constraints generally suggested intentionally codes, and provides international optimum solutions as critical native optimum solutions that square measure generally obtained with several alternative optimization strategies.

Jiangfeng Guo et al (2012) took some geometrical parameters of the shell and tube device because the style variables and therefore the genetic rule is applied to unravel the associated optimization drawback. it's shown that for the case that the warmth duty is given, not solely will the optimization style increase the heat money handler, effectiveness considerably, however additionally decrease the pumping power dramatically.

A. Piggott (2001) in his paper established relationship between the effectiveness of 2 heal money handler configurations that dissent from one another within the inversion of either one amongst 2 Studs. This paper provides the means by that if the effectiveness of 1 combination is thought in terms of warmth capability rate quantitative relation and NTUs, then the effectiveness of the opposite combination is pronto famous. Rajeev Mukharji (2017) explains the fundamentals of money handler thermal style, covering such topics as: parts, classification of STHES in line with construction and in line with service; information required for thermal style, tube facet design; shell facet design; as well as tube layout, baffling, and shell facet pressure drop; and mean temperature distinction. the essential equations for tube facet and shell facet heat transfer and pressure drop correlations for best condition also are targeted and explained with some tabulated information. This paper gives' overall ideal to style best shell and tube device. The best thermal style is done by refined pc code, but an honest understanding of the underlying principles of money handler styles is required to use this code effectively. Simin wangjian (2009) sebaceous cyst Yanzhong Li (2017) in his paper shows that the configuration of a shell and tube device was improved through the installation of sealers within the shell-side. The gaps between the baffle plates and shell are blocked by the sealers, that effectively decreases the contact flow within the shell-side. The results of warmth transfer experiments show that the shell-side heat transfer constant of the improved device exaggerated by eighteen.2-25.5%, the constant of warmth transfer exaggerated by fifteen.6-19.7% and therefore the potency exaggerated by 12-9-14.1%. Pressure losses exaggerated by forty four.6-48.8% with the sealer installation., however the increment of needed pump power is neglected compared with the increment of warmth flux. the warmth transfer performance of the improved device is intense, that is a visible profit to the optimizing of warmth money handler style for energy conservation.

Marner Bergles et al (1983) studied the hollow increased surface utilized in shell- and tube heat exchangers. As AN initial step, the topic is proscribed to single- section pressure drop and warmth transfer, however, each the tube facet and shell facet flows square measure taken into thought.

A comprehensive list of economic increased tubes which can be thought of to be used in shell and tube exchangers is given, in conjunction with a survey of the performance information that square measure out there within the literature. Ahmerrais Khan et al (2018), target the assorted researches on machine fluid dynamics (CFD) analysis within the field of warmth money handler. it's been found that CFD has been utilized for varied|the varied|the assorted} areas of study in various kinds of heat exchangers.

Different turbulence models offered normally purpose business CFD tools i.e standard, realizable and KNG k-£ RSM, and SST k-£ in conjunction with rate and pressure. Coupling schemes like easy, SIMPLEC, PISO and etc are adopted to hold out the simulations. The standard of the solutions obtained from these simulations square measure for the most part at intervals the suitable vary proving that CFDy is an efficient tool for predicting the behavior unhappy performance of a good form of heat exchangers. Philippe wildi- Tremblay (2007) in his paper explains the procedure for minimizing the value of a shell and tube device supported genetic algorithms (GA). The worldwide price includes the {operating price/operating expense/overhead/budget items/expense/disbursal/disbursement} (pumping power) and also the initial cost expressed in terms of annuities. He took some geometrical parameters of the shell and tube device because the style variables and also the genetic formula is applied to resolve the associated optimisation drawback. It's shown that for the case that the warmth duty is given, not solely will the optimisation style increase the warmth money changer effectiveness considerably, however conjointly decrease the pumping power dramatically.

Lunsford (1998) provided some ways for increasing shell and tube money changer performance. The ways thought-about whether or not the money changer is playacting properly to start with, excess pressured drop capability in existing exchangers, the re-evaluation of fouling factors and their impact on money changer calculations, and also the use of increased surfaces and increased heat transfer. Sparrow and Reifschneides (1986) conducted experiments on the impact of entomb baffle spacing on heat transfer. Huadong Li and Volker Kott ke (1998) conducted experiments on the impact of discharge on pressure drop native and native} heat transfer in shell and tube device for staggered has slight contribution to the local heat transfer at the surfaces of external tubes of the tube bundle, however reduced greatly the per compartment average heat transfer, Morcos Associate in Nursingd Shafey (1995) dispensed an experimental analysis to check the performance analysis of plastic shell and tube device.

Rozzi et al (2007) worked on convective heat transfer and friction losses in helically increased tubes for each Newtonian and non- Newtonian fluids. Four fluid foods, namely, whole milk, cloudy fruit crush, apricot and apple porea square measure tested in an exceedingly shell and tube device. Each fluid heating and cooling conditions square measure thought-about. The experimental outcome contains that helically furrowed tubes square measure significantly effective in enhancing convective heat transfer for generalized Reynolds range starting from regarding 800 to the limit of the transmutation flow regime. The heat exchanger effectiveness significantly, but also decrease the pumping power dramatically.

Lunsford (1998) provided some methods for increasing shell and tube exchanger performance. The methods considered whether the exchanger is performing correctly to begin with, excess pressured drop capacity in existing exchangers, the re-evaluation of fouling factors and their effect on exchanger calculations, and the use of augmented surfaces and enhanced heat transfer. Sparrow and Reifschneides (1986) conducted experiments on the effect of inter baffle spacing on heat transfer. Huadong Li and Volker Kott ke (1998) conducted experiments on the effect of leakage on pressure drop and local heat transfer in shell and tube heat exchanger for staggered has slight contribution to the local heat transfer at the surfaces of external tubes of the tube bundle, but reduced greatly the per compartment average heat transfer, Morcos and Shafey (1995) carried out an experimental analysis to study the performance analysis of plastic shell and tube heat exchanger.

Rozzi et al (2007) worked on convective heat transfer and friction losses in helically enhanced tubes for both Newtonian and non- Newtonian fluids. Four fluid foods, namely, whole milk, cloudy orange juice, apricot and apple porea are tested in a shell and tube heat exchanger. Both fluid heating and cooling conditions are considered. The experimental outcome contains that helically corrugated tubes are particularly effective in enhancing convective heat transfer for generalized Reynolds number ranging from about 800 to the limit of the transitional flow regime.

James Gosling et al 1991 designed the first version of java aimed at programming home appliances which are controlled by a wide variety of computer processors. His new language needed to be accessible by a variety of computer processors. In 1994, he realized that such a language would be ideal for use with web browsers and java's connection to the internet began. Java is a relatively new programming language. Netscape incorporated 1995 released its latest version of the Netscape browser which was capable of running java programs. It is customary for the creator of a programming language to name the language anything he/she chooses. The original name of this language was oak, until it was discovered that a programming language already existed that was named oak. As the story goes, after many hours of trying to come up with a new name, the development team went out for coffee and the name Java was born.

Java is viewed as a programming language to design applications for the internet. It is in reality a general all purpose language which can be used independent of the internet. The principles for creating java programming were simple, Robust, Portable, Platform independent, secured, high performance, multithreaded, Architecture neutral, object-oriented, interpreted and Dynamic. Currently, Java is used in internet programming, mobile devices, games, e-business solutions etc.

Software development is the process of convening, specifying, designing, programming, documenting, testing, and big fixing involved in creating and maintaining applications, frameworks, or other software components. Software development is a process of writing and maintaining the source code, but in a broader sense, it includes all that is involved between the conception of the desired software through to the final manifestation of the software, sometimes in a planned and structured process. Software development may include research, new development, prototyping, modification, reverse, re-engineering maintenance, or any other activities that result in software products.

Software can be developed for a variety of purposes, the three most common being to meet specific needs of a specific client/business (the case with custom software), to meet a perceived need of some set of potential users (the case with commercial and open source software) or for personal use. Embedded software development, that is, the development of the controlled physical product. System software underlies applications and the programming process itself, and is often developed separately. The need for better quality control of the software development process has given rise to the discipline of software engineering, which aims to apply the systematic approach exemplified in the engineering paradigm to the process of software development. There are many approaches to software project management, known as software development life cycle models, methodologies, processes, or models. The waterfall model is a traditional version, contrasted with the more recent innovation of agile software development.

A software development process (also known as a software development methodology, model, or life cycle) is a framework that is used to structure, plan, and control the process of developing information systems. There are several different approaches to software development, some take a more structured engineering-based approach to develop business more incremental approach, where software evolves as it is developed piece-by-piece. One system development methodology is not necessarily suitable for use by all projects. Each of the available methodologies is best suited to specific kinds of projects, based on various technical, organizational, project and team considerations.

Most methodologies share some combination of the following stages of software development: analyzing the problem, market research, gathering requirements for the proposed business solution, devising a plan or design for the software-based solution, implementation (coding) of the software, testing the software are:- deployment, maintenance and big fixing. These stages are often referred to collectively as the software development life cycle, or SDLC.

MATERIALS AND METHODS

Materials

Heat exchanger tubes and tube sheets can be constructed from any of the following materials (metals):

i. Admiralty: The serial number is 79006333 and admiralty metal tubes are used for heat exchangers in oil refineries, in which corrosion from sulfur compounds and contaminated water may be very severe and for feed- water heaters & heat exchanger equipment as well as other industrial processes
ii. Copper nickel: The serial number is 500164 and it is used for piping, heat exchangers and condensers in sea water systems as well as for marine hardware.
iii. Stainless steel: The serial number is 79237. It has higher levels of molybdenum, uranium etc which improve chloride pitting resistance and corrosion and thus has many heat transfer applications on the chemical and energy field.
iv. Low carbon steel: The serial number is 521345 and is used in making an anchor bolts, wire rod and metal products for various industries.
v. Hastalloy: Serial number is 8501590 and is extensively utilized in many industrial applications to exchanger heat from one place to another.

vi. Iconel: The serial number is 7145951 and is used to eliminate high cost down time which helps in offering the excellent corrosion resistance property in heat exchanger.

vii. Copper: The serial number is 603502 and it is used in industries to facilitate heat exchanger.

viii. Titanium: The serial number is 6009477 and is an excellent material for refrigerations equipment accessories.

Methods

The following mathematical models were used for the design of 1-2 vertical n- propanol condenser

(1) Kern's approach; It is mostly used for preliminary designs and provides conservative results. It is not cumbersome.

(i) The rate at which heat passed from the vapor through the liquid condensate film and into the cooling surface per unit area is given by

$$Q/A = k \frac{(t^1 - t)}{y^1} = \lambda W^1 = h(t^1 - t) \quad \text{..........} 3.1$$

where

λ is the latent heat of vaporization, W^1 is the pounds of condensate formed per hour per square foot and y^1 is the thickness of the condensate film at the generalized pint in the figure whose coordinated are x^1, y^1

The other symbols have their conventional meanings.

(ii) The rate at which vapor condensed is then given by: $W^1 = k \frac{(t^1 - t)}{\lambda y1}$ 3.2

(iii) The heat- transfer coefficient across the condensate layer at the distance x from the origin per unit of interfacial area is given by

$$hx = Qx/Ax = \frac{k/y^1}{(t^1 - t)} \quad \text{..........} 3.3$$

(vi) The total heat through the condensate layer from O to X is Q_x

$$Qx = \int_0^x h_x\, {}^1 - t)\; dx = \int_0^x k \left(\frac{p^2 \lambda g}{4\mu(t^1 - t)} \right)^{\frac{1}{4}} \frac{dx}{x^{1/4}}$$

$$= \frac{4}{3}\, (k32\lambda g)1/4\; \left[\frac{(t^1 - t)x}{\mu}\right]^{3/4} \quad \text{..........} 3.4$$

(v) When the liquid descends vertically on the outside of a tube it is certainly in streamline flow at the top of the tube, where the accumulation of condensate is small. In the case of the double pipe exchange, the equivalent diameter was take as four times the hydraulic radius

(vi) Then De = $4r_h$ = 4 x free $\frac{flow\,ara}{wetted\ perimeter}$

$$Re = \frac{D_e G}{\mu}$$

For vertical tubes let

A_f = cross sectional area (shaded)

P = wetted perimeter per tube

$D_e = 4 \times A_f/p$

Letting the loading per tube be $w^1 = w/Nt$, where Nt is the number of tubes

$G = w^{-1}/A_f l_b/$ (hr) (ft^2)

$R_e = D_e\, G/\mu = (4A_f/p)\,(w^1/A_f)/\mu = 4W^1/\mu P$

Calling the condensate loading per linear foot G^1

$$G^1 = \frac{W^1}{P^1} \text{ lb/(hr) (lin ft)}$$

Equation (3.1) becomes $= \dfrac{R_e G^1}{\mu}$

The total heat load is given: $Q = \lambda w$

$$h^1 = \frac{Q}{A\Delta tf} = \frac{\lambda w1}{PL\Delta tf} = \frac{\lambda G^1}{L\Delta tf} \quad \text{...} \quad 3.5$$

$$h = 0.943 \left(\frac{}{\mu} \quad \frac{}{G^1} \right) \quad kf^3pf \quad 2g \;\; h$$

Multiplying right term by $(4\mu/0.4\mu)1/4$

$h^{-}3/4 = 0.943\; 4k$

$$\left(\frac{4k_f^3\rho_f^2 g}{\mu ff} \quad \frac{\mu f}{4G^1} \right) \quad \text{..} \quad 3.6$$

$$h \left(\frac{\mu_f^2}{k_f^2\rho_f^2 g} \right) 1/3 = 1.47 \left(\frac{4G^1}{\mu f} - 1/3 \right)$$

For vertical $h = \dfrac{4000}{L^{1/4}\,\Delta t_f^{\;1/3}}$

where Δt_f ranges from 10 to 150^{0}F

The hydraulic circuit consists of the weight of the over heart vapor line Z_3- Z_2, condenser pressure drop ΔPc, the weight of the condensate leg Z_1, the pressure in the condensate return line which is usually neglected. The equation is given closely in pounds per square inch by:

$$\frac{PVZ1}{144} + \Delta Pc = \frac{PLZ1}{144} \quad \ldots\ldots\ldots 3.7$$

Where P_v = density of vapour Lb/ft^3

P_L = density of liquid Lb/ft^3

$^{\Delta}Pc$ = pressure drop in condenser, Ps,

∴ For condensation in the shell,

$$\Delta P_s = \frac{1}{2}\frac{FG_S^2 D_s\,(N+1)}{5.22x10^{10}\ D_e S} \quad \ldots\ldots\ldots 3.8$$

where S is the Specific gravity of vapour. For condensation in tubes:

$$\Delta p_t = \frac{1}{2}\frac{FG_t^2 L_n}{5.22x10^{10}\ D_e S} \quad \ldots\ldots\ldots 3.9$$

where S is the Specific gravity of vapour. No contraction or expansion losses need be considered.

The shell side or bundle cross flow area as is then given by $\propto \ = \ \frac{DXC^1 B\ \ Ft^2}{P_T\ X\ 144\ P_T}$

And as before, the mass velocity is

$G_s = \frac{W}{\alpha s}$ lb/(hr) (ft^2) (Mc Adams. W.H) 1942

for triangular pitch, the wetted perimeter of the element corresponds to half a tube

$$d_e = \frac{4\ x\ (1/2P_T\ X\ 0.86P_T - ½\ \Pi\ d_0^2/4)in}{½\ \Pi do} \quad \ldots\ldots\ldots 3.10$$

∴The overall heat balance, where Δt is the true temperature difference is Q =UAΔt = WC $(T_1\text{-}t_2)$ = WC $(t_2\text{-}t_1)$ from which Δ_t

$$= \left(\frac{T1 - T2}{UA/Wc}\right)_{true} = \left(\frac{t^1 - t^2}{UA/Wc}\right)_{true}$$

The fractional ratio of the true temperature difference to the LMTD Fr.

$$F_r = \frac{\sqrt{R^2+1}\ \ln\ (\ \ S)/(1\text{-}RS)}{(R\text{-}1)\ \ln\frac{2-S(R+1-\sqrt{R^2+1}}{2-S(R+1-\sqrt{R^2+1}}} \quad \ldots\ldots\ldots 3.11$$

∴The Fourier equation for a 1-2 exchanger is $Q = UA\Delta t = UAF_T\,(LMTD)$ (Fourier)
∴The isothermal equation for the pressure drop of a fluid being heated or cooled and including entrance and exit losses is

$$\Delta Ps = \frac{FG_S^2 D\,(N+1)}{2gp\,De\varphi s} = \frac{FG_S^2 DS\,(N+1)}{5.22 \times 10^{10} De\,S\varphi s} \quad \ldots\ldots 3.12$$

Sieder and Tate have correlated friction factors for fluids being heated or cooled in tubes. They are plotted in dimensional form and are used in the equation.

$$\Delta P_t = \frac{FG_t^2 L_n}{5.22\mathrm{x}10^{10}\,D_e S\phi_t}\ \mathrm{Psf} \quad \ldots\ldots 3.13$$

where n is the number of tube passes, L the tube length, and Ln is the total length of path in feet. The deviations are not given, but the curve has been accepted by the Tubular Exchanger Manufacturers Association. In flowing from one pass into the next at the channel and floating head the fluid changes direction abruptly by 180^0, although the flow area provided in the channel and floating-head cover should not be less than the combined flow area of all the tubes in a single pass. the change of direction introduces an additional pressure drop ΔPr, called the return loss and accounted for by allowing four velocity heads per pass. The velocity head $V^2/2g^1$ has been plotted against the mass velocity for a fluid with a specific gravity of 1, and the return losses for any fluid will be

$$\Delta p_r = \frac{4n}{s}\frac{V^2}{2g^1}\mathrm{Psi} \quad \ldots\ldots 3.14$$

Where V = velocity, fps
s = specific gravity
g^1 = acceleration of gravity, ft/sec^2
The total tube-side pressure drop ΔP_T will be

$$\Delta P_T = \Delta P_t + \Delta P_r\ \mathrm{psi} \quad \ldots\ldots 3.15$$

Numerical example: calculation of a vertical n- propanol condenser. 2.52kg/s of a substantially pure n-propanol (propyl alcohol) coming from the top of a distilling column operating at 103421.35n/m^2 at which pressure it boils at 390.93k is to be condensed by water from 302.59k to 322.04k. A dirt factor of 0.000529m^2k/w is required with allowable pressure drops of 13789.52n/m^2 for the vapour and 68947.53n/m^2 for the water. Determine the size of a 0.79m ID 1-2 vertical condenser with 0.019m OD, 16 BWG tube on 0.0238m triangular pitch which is required for this service.
Solution
1) Heat balance:
n- propanol, Q = 2.52 x 662910 = 1670533.2J/S
water, Q = 20.45X4200 (322.04- 302.59)
= 1670533.2J/S

Table 3.1: Temperature Differences between hot and cold fluid

Hot fluid (n-propanol)		Cold fluid (Water)	Temperature difference between hot fluid and cold fluid
390.93K	Higher temperature	322.04K	68.89K
390.93K	Lower temperature	302.59K	88.34K
OK	Difference	19.45K	19.45K

Δt = LMTD = 78.21K

The exchanger is in true counter flow, since the shell side fluid is isothermal

3. Tc and t_c: The influence of the tube wall temperature is included in the condensing film coefficient. The mean t_∞= 312.32k can be used for t_c in condensation calculations, the omission of the resistance of the tube metal may introduce a significant error and should be checked.

a) assume U_D = 397.35J/Sm^{2}k. The equation for the condensing film coefficient gives greater values for horizontal tubes than for vertical tubes. It will be consequently be necessary to reduce the valve of U_D

$$A = \frac{Q}{UD\,\Delta t} = \frac{1670533.2}{93.44 \times 78.21} = 53.755m^2 \quad \ldots \quad (3.16)$$

$$\text{Common tube length} = \frac{53.755}{766 \times 0.0599} = 1.172m$$

b) The layout of the example using 0.019m OD tube on 0.0238m triangular pitch and four passes will be retained for comparison. The quantity of water is large but the condenser will have a large number of tubes, making a two-pass assumption inadvisable.

c) Corrected coefficient U_D

A = 766 x 1.172 x 0.0599 = 53.775m^2

$$U_D = \frac{Q}{A\Delta t} = \frac{1670533.2}{53.775 \times 78.21} = 397.203J/sm^2k \quad \ldots \quad (3.17)$$

BeIl's method

In Bell's method, the heat transfer coefficient and pressure drop are estimated from correlations for flow over ideal tube- banks, and the effects of leakage, by passing and flow in the window zone are allowed for by applying correction factors. This approach takes into account the effects of leakage and by passing, can be used to investigate the effects of constructional tolerances and the use of sealing strips. The procedure in a simplified and modified form to that given by Bell (1963), is outlined below.

The method is not recommended when the bypass flow area is greater than 30% of lie cross flow area, unless sealing strips- are used,

Heat transfer coefficient.

The shell side heat transfer coefficient is given by: $h_s = h_{oc} F_n\, F_w\, F_b FL$

where h_{oc}= heat transfer coefficient calculated for cross- flow over an ideal tube bank, no leakage or by passing.

F_n = correction factor to allow for the effect of the number of vertical tube rows.

F_w = window effect correction factor

F_b = bypass stream correction factor

F_L = Leakage correction factor

The total correction will vary from 0.6 for a poorly designed exchanger with large clearances to 0.9 for a well-designed exchanger.

$h_{OC,}$ ideal cross-flow coefficient

the heat-transfer coefficient for an ideal cross- flow tube bank can be calculated using the heat transfer factors. The comprehensive data given in the Engineering Sciences Data unit. Design Guide on heat transfer during cross- flow of fluids over tube banks, ESDU 73031 (1973) can be used; see Bitter worth (1977).

$$R_e = \frac{GsDo}{\mu} = \frac{\mu s\, \rho dc}{\mu} \ldots \quad 3.20$$

The Reynolds number for cross- flow through a tube banlc is given by:

Where G_s = mass flow rate per unit area, based on the total flow and free are at the bundle equator. This is the same as G_s calculated for kern's method.

do = tube outside coefficient is given by

$$\frac{h\alpha}{kf} = J_h\, R_e\, P_r\, 1/3 \quad \left(\frac{\mu}{\mu w}\right)^{0.14} \quad \ldots \quad 3.21$$

F_n; tube row correction factor

The mean heat transfer coefficient will depend on the number of tubes crossed. For turbulent flow the correction factor F_n is close to 10. in laminar flow, the heat transfer coefficient may decrease with increasing rows of tubes crossed, due to the build up of the temperature boundary layer. The factors given below can be used for the various flow regimes "the factors for turbulent flow are based on those given by Bell (1963) N_{cv} is number of constructions crossed = number of tube rows between the baffle tips;

i) Re >2000, turbulent
ii) Re > 100 to 2000, transition region
iii) Re < 100, laminar region.

$F_n \propto (N_C^1)^{-0.18}$

Where N^1_c is the number of rows crossed in series from end to end of the shell, and depends on the number of baffles. The correction factor in the Laminar region 'is not well established, and Bell's paper or the summary given by Mueller (1973), should be consulted if the design falls in this region.

F_W, window correction factor

This factor corrects for the effect of flow through the baffle window and is a function of the heat transfer area in the window zones and the total heat-transfer area.

F_b, bypass correction factor.

This factor corrects for the main bypass stream, the flow between the tube bundle and the shell wall,and is a function of the shell to bundle clearance, and whether sealing strips are used.

$$F_b = \exp\left[\frac{-\propto A_b}{A_s}\left(1-\left(\frac{2N_s}{N_{cv}}\right)^{1/3}\right)\right] \quad \text{3.22}$$

Where $\propto$= 1.5 for laminar flow, Re< 100

A = 1.32 for transitional and turbulent flow Re > 100

A_b = clearance area between the bundle and the shell

As = Maximum area for cross- flow

Ns = number of sealing strips encountered by the by pass stream in the cross-flow zone.

N_{cv} = the number of constrictions, tube rows, encountered in the cross-flow section.

F_L, leakage correction factor

This factor corrects for the leakage through the tube- to- baffle clearance and the baffle -to- shell clearance

$$F_L = 1-B_L\left(\frac{A_{tb} + 2A_{sb}}{A_L}\right) \quad \text{3.23}$$

Where B_L = a factor

A_{tb} = the tube to baffle clearance, per baffle

A_{sb} = shell -to- baffle clearance area, per baffle

A_L = total leakage area = $(A_{tb} + A_{sb})$

The clearances and tolerances required in practical exchangers are discussed by Rubin (1968).

Pressure drop:

The pressure drops in the cross- flow and window zones are determined separately, and summed to give the total shell- side pressure drop.

Cross- flow zones:

The pressure drop in the cross- flow zones between the baffle tips is calculated from correlations for ideal tube banks and corrected for leakage and by passing.

$\Delta_{pc} = \Delta_{pi} F_b^1 F_L^1$

Where Δ_{pc} = pressure drop in a cross- flow zone between the baffle tips, corrected for by- passing and leakage.

Δ_{pc} = the pressure drop calculated for an equivalent ideal tube bank.

F_b^1= bypass correction factor

F_L^1 = leakage correction factor.

Δ_{pi} ideal tube bank pressure drop

The number of tube rows has little effect on the friction factor and is ignored. The pressure drop across the ideal tube bank is given by:

$$\Delta pi = 8j_f\ N_{cv} \frac{\rho u_s^2}{2} \left(\frac{\mu}{\mu w}\right)^{-0.14} \quad \text{...3.24}$$

Where N_{cv}= number of tube rows crossed (in the cross - flow region).
U_s = shell side velocity, based on the clearance area at the bundle equator,
J_f = friction factor at the appropriate Reynolds number, Re = $(\rho u_s do/\mu)$
Fb^1, bypass' correction factor for pressure drop bypassing will affect the pressure drop only in the cross-flow zones. The correction factor is calculated from the equation used to calculate the bypass. Correction factor for heat transfer, but with the following values for the constant a. Laminar region, Re < 100, $\propto$= 5.0
Transition and turbulent region, Re > 100, $\propto$= 4.0
F_L' leakage factor for pressure drop leakages will affect the pressure drop in both the cross-flow and window zones. The factor is calculated using the equation for the heat transfer leakage - correction factor with the values for the coefficient BL' taken from the figure.
Window - zone pressure drop
Any suitable method can be used to determine the pressure drop in the window area: see Butter worth (1977). Bell used a method proposed by colburn. Corrected for leakage, the window drop for turbulent flow is given by:

$$\nabla p_w = F_L^1 \ (2+0.6\ N_{wv}) \frac{\rho_{\mu z}^2}{2} \quad \text{... 3.25}$$

Where U_z = the geometric mean velocity

$U_Z = \sqrt{Uw \quad Us}$

U_W = the velocity in the window zone, based on the window area less the area occupied by the tubes Aw

$U_w = \frac{W_s}{A_w \rho}$

Ws = shell side fluid mass flow, kg/s

Nwv = number of restrictions for cross- flow in window zone, approximately equal to the number of tube rows.

End zone pressure drop

There will be no leakage paths in an end zone (the zone between tube sheet and baffle). Also, there will only be one baffle window in these zone; so the total number of restrictions in the cross-flow will be N_{cv} + $N_{wv.}$

The end zone pressure drop Δp_e= $\left(\frac{N_{wv} + N_{cv.}\ F^{1}_{b}}{N_{cv}}\right)$ 3.26

Total shell-side pressure drop

Summing the pressure drops over all the zones in series from inlet to outlet gives;

Δ_{ps} = 2 end zones + (N_b + 1) cross flow zones + N_b window zones

$\Delta_{ps} = 2\ \Delta p_e + \Delta_{pc}\ (N_b\ -1) + N_b\Delta_{pw}$

Where N_b is the number of baffles = [(L/L_B)-1]

Shell and bundle geometry.

$H_b = \frac{Db}{2} = D_s\ (0.5\text{-}B_c)$

$N_{cv}\ \frac{(D_6 - 2H_b)}{P^1_t}$ 3.27

Where H_c = baffle cut height = D_s x B_c, where

Bc is the baffle cut as a fraction,

H_b = height from the baffle chord to the top of the tube bundle

B_b = "bundle cut" = H_b/D_b

θ_b = angle subtended by the baffle chord, rads.

D_s = bundle diameter

Matlab Program Design

The design of this project in MATLAB Programming Language is dominantly a procedural design. It is made up of 8 major scripts which are: Condensate Loading, Condensing Film, Dirt Factor, Mass Velocity, Outside Diameter, Pressure Drop, Reynolds Diameter and Wall Temperature

Each script is written in a top-down format. The steps are:

1. The script waits for inputs by the user. These inputs are then defined as a variable used in program. Execution of the script will only continue if the user types in a value for all the inputs.
2. Constants are also defined in the program. Note that these constants cannot be changed by the user during execution of the script.
3. The variables and constants are used in the algorithm for calculating the right result.
4. After calculation is done, the results are then printed out to the screen so that the user can see result.

Since MATLAB is a weakly typed language, checks are carried out to make sure that the inputs are of the correct data types i.e integers are checked to make sure that they are integers and floats are checked to make sure they are floats and also to make sure they are in the correct range.

There are no functions defined in this design to take advantage of the speed of iteration in designing this program. No external libraries or functions are used in the course of creating this project. All functions used are internal ones embedded in the MATLAB language.

This project is designed as an interactive one in which the user interacts with the program using their command line shell.

Java Program Design

The design of this project in Java Programming Language is dominantly an Object-oriented design. It is made up of 9 major components which are: Main Component, Condensate Loading, Condensing Film, Dirt Factor, Mass Velocity, Outside Diameter, Pressure Drop, Reynolds Diameter, Wall Temperature

Each component is divided into two parts: User Interface and Controller

Each component's **user interface** is written in an xml file. The reason is that in the Java program, the JavaFX UI library is used. Java FX recommends that the structure of the user interface be written in xml specifically **fxml.** XML is a markup language that is made up of tags and attributes. In fxml, different layouts and user interface components are declared in opening and closing tags. For example, a button is declared like this "<Button text='calculate'></Button>". XML attributes are used for defining the properties of these layouts and components like height, width, text etc. Each xml file is linked to the component controller. This is to ensure the relationship between the user interface and the right controller. Also, the button tag has an attribute `on Action` which has to be a function in the controller that executes an action when it is clicked.

The **controller** is written in Java. Each controller is java file where one class is defined. In this class, the ui components are declared as a field. Every component has at least one function that is linked to Button component. This function executes when the linked button is clicked on. When this function executes, the following steps are taken:

1. The inputs from the ui components field are initialized to a variable and converted to their correct data types
2. Then the calculations are carried out in their respective functions which are also defined as a method in the class. These functions accept the inputs as parameters and then execute a series of algorithmic steps. The final result is returned and stored as a variable.
3. This final result is then printed out to the correct ui components so that the user can view the results.

Checks are carried out through out these steps to make sure that there are no errors in the final results.

In summary the design of this program is a View-Controller design.

RESULTS

The effects of the design variables on the heat loads were shown in the table below

Table 1: Heat transfer surface vs heat load

Heat transfer surface (M^2)	Heat Load (J/S)
53.775	1670533.2
107.505	3341066.4
161.280	5011599.6
215.010	6682132.8
268.877	8352666.0

Table 2: Actual condensing film coefficient vs heat load

Actual condensing film coefficient j/sm2k	Heat Load (J/S)
397.203	1670533.2
397.369	3341066.4
397.314	5011599.6
397.370	6682132.8
397.200	8352666.0

Table 3: Condensate loading of vertical tube vs heat load

Condensate loading of vertical tube (kg/sm)	Heat Load (J/S)
0.055	1670533.2
0.110	3341066.4
0.165	5011599.6
0.221	668212.8
0.276	8352666.0

Table 4: Mass velocity of shell vs heat load

Mass velocity of shell vs heat load{kg/sm2}	Heat Load (J/S)
68.007	1670533.2
68.036	3341066.4
68.027	5011599.6
68.108	6682132.8
68.035	8352666.0

Table 5: Reynold number of tube vs heat load

Reynold number of tube	Heat Load (J/S)
12,000.000	170533.2
24,000.000	3341066.4
36,000.000	5011599.6
48,000.000	6682132.8
60,000.000	8352666.0

Table 6: Flow area of shell vs heat load

Flow area of shell (M^2)	Heat Load (J/S)
0.0371	1670533.2
0.0741	3341065.4
0.1111	5011599.6
0.1480	6682132.8
0.1852	8352666.0

Table 7: Pressure drop in shell vs heat load

Pressure drop in shell N/M^2	Heat Load (J/S)
17236.890	1670533.2
17236.890	3341066.4
17236.890	5011599.6
17236.890	6682132.8
17236.890	8352666.0

Table 8: Total pressure drop in tube vs heat load

Total pressure drop in tube N/M^2	Heat Load (J/S)
4103.635	1670533.2
21110.125	3341065.4
54995.83	5011599.6
107006.635	6682132.8
185744.766	8352666.0

Table 9: Baffle space vs heat load

Baffle space (M)	Heat Load (J/S)
0.234	1670533.2
0.469	3341066.4
0.703	5011599.6
0.937	6682132.8
1.172	8352666.0

Table 10: Mass velocity of tube Vs heat load

Mass velocity of tube kg/sm^2	Heat Load (J/S)
547.633	1670533.2
1095.267	3341066.4
1642.900	5011599.6
2190.534	6682132.8
2738.167	8352666.0

Table 11: Velocity of water flowing in tube vs heat load

Velocity of water flowing in the tube (M/S)	Heat Load (J/S)
0.547	1670533.2
1.094	3341066.4
1.641	5011599.6
2.188	6682132.8
2.735	8352666.0

Table 12: Tube length vs heat load

Tube length (m)	**Heat Load (J/S)**
1.172	1670533.2
2.343	3341066.4
3.515	5011599.6
4.686	6682132.8
5.858	8352666.0

Table 13: Reynold number of shell vs heat load

Reynold number of shell	**Heat Load (J/S)**
95089.000	1670533.2
95200.000	3341066.4
95116.000	5011599.6
95230.000	6682132.8
95127.000	8352666.0

Table 14: Temperature of tube wall vs heat load

Temperature of tube wall K	**Heat Load (J/S)**
326.490	1670533.2
321.120	3341066.4
319.013	5011599.6
317.540	6682132.8
316.800	8352666.0

Table 15: Mass flow rate of n-propanol vs heat load

Mass flow rate of n-propanol kg/s	**Heat Load (J/S)**
2.520	1670533.2
5.040	3341066.4
7.560	5011599.6
10.080	6682132.8
12.600	8352666.0

Table 16: Temperature of film vs heat load

Temperature of film (k)	**Heat Load (J/S)**
358.710	1670533.2
356.025	3341066.4
354.972	5011599.6
354.235	6682132.8
353.865	8352666.0

Table 17: Dirt factor vs heat load

Dirt factor m^2k/w	**Heat Load (J/S)**
0.000986	1670533.2
0.000854	3341066.4
0.000625	5011599.6
0.000493	6682132.8
0.000366	8352666.0

DISCUSSION

From fig 1, increase in heat transfer surface increases the heat load and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is a linear graph (positive slope)

From fig 2; when the heat load increases, the actual condensing film coefficient increases, then decreases, increases again and finally decreases at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water,

tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is not a linear graph.
From fig 3; when the heat load increases, the condensate loading of vertical tube increases and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is linear graph (positive slope).
From fig 4: when the heat load increases, the mass velocity of shell increases, then decreases, increases again and finally decreases at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So it is non linear graph.
From fig 5: when the heat load increases, the Reynold number of tube increases and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So it is a linear graph. (positive slope)
From fig 6:As the heat load increases so does the flow area of shell increases and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So it is a linear graph. (positive slope)
Fig 7: As the heat load increases, the pressure drops in shell remain constant, at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol.
Fig 8: As heat load increases, the total pressure drop increases and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is a curve.
Fig 9: As heat load increases, the baffle space increases linearly and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is linear.
Fig 10: As heat load increases, the mass velocity of tube increases linearly and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is linear.
Fig 11: As heat load increases, so does the velocity of water flowing in the tube increases linearly and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is linear. (positive slope)
Fig 12: As heat load increases, so does the tube length increases and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is linear (positive slope).
Fig 13: As heat load increases, the Reynold number of shell increases to a point, then decreases, increases again and finally decreases at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is not linear.
Fig 14: As heat load increases, the temperature of tube wall decreases at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So it is a curve

Fig 15: when the heat load increases, the mass flow rate of n-propanol increases linearly and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol. So the graph is linear (positive slope).
Fig 16: when the heat load increases, the temperature of film decreases versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol.
Fig 17: when the heat load increases, the dirt factor decreases and vice versa at constant logarithmic mean temperature difference, constant flow area of tube, number of tubes, tube pitch, tube outside diameter, shell inside diameter, density of water, tube inside diameter, shell outside diameter, boiling point of n-propanol.

CONCLUSION
From 1st heat load, the condenser is somewhat safe from a heat transfer standpoint but exceeds the allowable pressure drop, although not seriously. The calculated dirt factor exceeds the required dirt factor and this will warrant the installation of the unit. The exchanger is satisfactory for the service.
From 2nd heat load, the condenser is somewhat safe from a heat transfer stand point but exceeds the allowable pressure drop although not seriously. The calculated dirt factor exceeds the required dirt factor and this will warrant the installation of the unit. The exchanger is satisfactory for the service.
From 3rd heat load, the condenser is somewhat safe from a heat transfer stand point but exceeds the allowable pressure drop although not seriously. . The calculated dirt factor exceeds the required dirt factor and this will warrant the installation of the unit. The exchanger is also satisfactory for the service.
From 4th heat load, the condenser is not safe from a heat transfer stand point because it exceeds the allowable pressure drop seriously. The calculated dirt factor is less than the required dirt factor and this will not warrant the installation of the unit. The exchanger is not satisfactory for the service.
From 5th heat load, the condenser is not safe from a heat transfer stand point because it exceeds the allowable pressure drop seriously. The calculated dirt factor is too small to warrant the installation of the unit. The exchanger is not satisfactory for the service.

5.3 SUMMARY

1)	For 1st heat load
	874.490 h outside 2579.800
	Uc 653.100
	UD 397.200
	Rd calculated 0.000986
	Rd required 0.000529
	17236.890 calculated Δp 4103.635
	13789.515 allowable Δp 68947.573
Shell side	Tube side
ID = 0.79m	number and length of tube = 766,1.172m
Baffle space = 0.234m	OD, BWG, pitch 0.019m, 16,0.0238
Passes = 1	passes = 4
2)	for 2nd heat load
	694.100 h outside 4502.950
	Uc 601.400
	UD 397.370
	Rd calculated 0.000854
	Rd required 0.000529

17236.890 calculated Δp 21110.12
13789.515 allowable Δp 68947.573

Shell side	Tube side
ID = 0.79m	number and length of tube = 766,2.343
Baffle space = 0.469m	OD, BWG, pitch 0.019m, 16,0.0238
Passes = 1	passes = 4

3) For 3rd heat load
578.800 h outside 6097.740
Uc 528.62
UD 397.314
Rd calculated 0.000625
Rd required 0.000529
17236.89 0 calculated Δp 54995.83
13789.515 allowable Δp 68947.573

Shell side	Tube side
ID = 0.79m	number and length of tube = 766,3.515
Baffle space = 2.99m	OD, BWG, pitch 0.019m, 16,0.0238
Passes = 1	passes = 4

For 4th heat load
4) 526.82 h outside 7973.970
Uc 494.170
UD 397.370
Rd calculated 0.000493
Rd required 0.000529
17236.89 calculated Δp 107006.633
13789.515 allowable Δp 68947.573

Shell side	Tube side
ID = 0.79m	number and length of tube = 766,4.686
Baffle space = 0.937m	OD, BWG, pitch 0.019m, 16,0.0238
Passes = 1	passes = 4

5) For 5th heat load
489.080 h outside 9381.143
Uc 464.850
UD 397.200
Rd calculated 0.000366
Rd required 0.000529
17236.890 calculated Δp 185744.766
13789.515 allowable Δp 68947.573

Shell side	tube side
ID = 0.79m	number and length of tube = 766,5.858
baffle space = 1.172m	OD, BWG, pitch 0.014 m, 16, 0.0238
Passes = 1	passes = 4

RECOMMENDATIONS

- The oil chamber of the money handler might become crammed with sludge accumulation and need cleanup. it's suggested that the unit be flooded with a billboard solvent and left to soak for one – 0.5 hour.
- Back flowing with the solvent or regular oil can take away most sludge. Perennial soaking and back flowing is also needed relying on the degree of sludge build up.

- It may be necessary to wash the within of the cooling tubes to get rid of any contamination and/or scale build up. It's suggested that a fifty- half resolution of repressed hydrochloric acid and water is also used. For severe issues, the use of brush through the tubes is also of some facilitate. Make certain to use a soft bristly brush to forestall scouring the tube Surface inflicting accelerated corrosion. Upon completion of cleanup, be sure that all chemicals area unit removed from the shell aspect and also the tube aspect before the warmth exchange is placed in commission.
- once ordering replacement components or creating associate degree inquiry relating to service, mention model variety, serial variety and also the original purchase order variety.
- To prevent corrosion of the shell, it's suggested that water can flow within the tube.

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