Fixed Equipment Newsletter

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Systematic Procedure for Selection of Heat Exchangers (p. 7)

Serving the Pressure Vessel Community Since 2007

From The Editor's Desk:

With this issue, we are changing the name of the newsletter to "Fixed Equipment Newsletter" to more accurately reflect the type of articles we have been publishing. In addition to pressure vessels, the newsletter since its beginning has published articles on heat exchangers and storage tanks as well; and going forward, plans to continue writing articles about all fixed equipment.

Also, the articles will not be limited to the design of process equipment. My own personal involvement has overwhelmingly been in the design area; however, every effort will be made to address other aspects of process equipment also – both pre-construction and post-construction. If you would like to contribute to the newsletter, please let us know at the email address provided below.

As far as design is concerned, the articles will mainly focus on the mechanical design of equipment and not the process design. Finite element analysis is increasing being used not only for the design of the new equipment but also for the evaluation of existing equipment. Future articles will undoubtedly have to address these "non-code" analysis of the equipment as well.

And as always, we thank you very much for the important feedback that you provide on the format and the content of the newsletter.

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THE GRAPHIC ON THE COVER PAGE HAS BEEN PROVIDED BY FOURINOX IN GREEN BAY, WISCONSIN.

In this issue…

HYDROSTATIC TEST OF PRESSURE VESSELS

Why is the pressure test performed?

The reasons for performing pressure test on pressure vessels is not stated in the ASME Section VIII, Division 1. The most obviously of reasons is that the pressure test serves as a proof of design and workmanship verification. However, the major reasons for performing the pressure test are as follows:

- The pressure test uncovers gross errors, due to design or workmanship, including leaks at welded or flanged connections.
- The application of the test pressure results in a stress relief of the vessel, where local areas of high stress, either due to design or fabrication issues, undergo local yielding at the test pressure, resulting in a better stress pattern after release of the pressure.

Regardless of the reasons behind the pressure test, the test must always be performed after all other fabrication steps are complete.

When is hydrostatic test conducted on pressure vessels?

Hydrostatic test is conducted after all fabrication has been completed and all examinations have been performed. Exceptions are those operations that could not be performed prior to the test such as weld end preparation and any cosmetic grinding on the base material that does not reduce the actual thickness below the design thickness.

The requirements for hydrostatic testing of pressure vessels are provided in paragraph UG-99 of ASME Section VIII, Division 1.

How is maximum allowable work pressure (MAWP) defined?

According to ASME Section VIII, Division 1 (ASME VIII-1), MAWP is the maximum gage pressure at the top of a completed vessel in its normal operating position at the designated coincident temperature for that pressure. This pressure is the least of the values for the internal or external pressure to be determined by the rules of ASME VIII-1 for any of the pressure boundary parts, including the static head thereon, using nominal thickness exclusive of allowances for corrosion and considering the effects of any combination of loadings listed in UG-22 that are likely to occur at the designated coincident temperature. It is the basis for pressure setting of the pressure relieving devices protecting the pressure vessel. The design pressure may be used in all cases in which calculations are not made to determine the value of the MAWP.

What is "Lowest Stress Ratio" used in the calculation of the minimum hydrostatic test pressure of a pressure vessel?

Lowest Stress Ratio (LSR) is the smallest ratio of the allowable stress at test temperature to the allowable stress at design temperatures of materials used in the vessel construction. (Bolting is excluded except when the calculated test pressure will exceed 90% of the bolt material minimum yield strength at the test temperature.)

What are "Authorized Inspection Agencies"?

An Authorized Inspection Agency (AIA) is an organization that meets the criteria of the ASME QAI-1 standard, "Qualifications for Authorized Inspection" and that is accredited by the ASME to provide AIA services. Authorized Inspection Agencies include:

- Jurisdictional Authorities: A jurisdiction that accepts and administers ASME VIII-1 as a means to satisfy legal or regulatory requirements.
- Insurance Companies: An insurance company which has been licensed by the appropriate authority of a state of the United States of America, or of a province of Canada, to write boiler and pressure vessel insurance in such state or province.
- Independent Third-Party Inspection Organizations: A company in the business of providing third party inspection services, which has government recognition to perform inspection and design reviews for boilers and pressure vessels.

Who is an "Authorized Inspector"?

The individuals performing authorized inspections under the employment of an accredited organization are called Authorized Inspectors.

How is the minimum hydrostatic test pressure determined?

Pressure vessels designed for internal pressure shall be subjected to a hydrostatic test pressure that at every point in the vessel is at least equal to 1.3 times the maximum allowable working pressure multiplied by the lowest stress ratio (LSR) for the pressure-boundary materials of which the vessel is constructed. All loadings that may exist during this test shall be given consideration. The hydrostatic test pressure reading shall be adjusted to account for any static head conditions depending on the difference in elevation between the chamber being tested and the pressure gauge.

ASME VIII-1 does not specify an upper limit for hydrostatic test pressure. However, if the hydrostatic test pressure is allowed to exceed, either intentionally or accidentally, the minimum value determined as prescribed above to the degree that the vessel is subjected to visible permanent distortion, the Authorized Inspector (AI) shall reserve the right to reject the vessel.

What is a Combination Unit?

A combination unit is a pressure vessel that consists of more than one independent or dependent pressure chamber, operating at the same or different pressures and temperatures. The parts separating each pressure chamber are the common elements. Each element, including the common elements, shall be designed for at least the most severe condition of coincident pressure and temperature expected in normal operation.

How are pressure vessels prepared before the hydrotest?

Vents shall be provided at all high points of the vessel in the position in which it is to be tested to purge possible air pockets while the vessel is filling. Before applying pressure, the test equipment shall be examined to see that it is tight and that all low pressure filling lines and other appurtenances that should not be subjected to the test pressure have been disconnected.

How are Combination Units hydrotested?

Combination units shall be tested by one of the following methods:

- Independent Pressure Chambers: Pressure chambers of combination units that have been designed to operate independently shall be hydrostatically tested as separate vessels, that is, each chamber shall be tested without pressure in the adjacent chamber. If the common elements of a combination unit are designed for a larger differential pressure than the higher maximum allowable working pressure to be marked on the adjacent chambers, the hydrostatic test shall subject the common elements to at least their design differential pressure, corrected for temperature, as well as meet the requirements for each independent chamber.

- Dependent Pressure Chambers: When pressure chambers of combination units have their common elements designed for the maximum differential pressure that can possibly occur during startup, operation, and shutdown, and the differential pressure is less than the higher pressure in the adjacent chambers, the common elements shall be subjected to a hydrostatic test pressure of at least 1.3 times the differential pressure to be marked on the unit, corrected for temperature.

Following the test of the common elements and their inspection, the adjacent chambers shall be hydrostatically tested simultaneously. Care must be taken to limit the differential pressure between the chambers to the pressure used when testing the common elements.

How are pressure vessels designed for vacuum only tested?

Single-wall vessels and individual pressure chambers of combination units designed for vacuum only (MAWP less than or equal to zero) shall be subjected to either:

- An internal hydrostatic pressure test or a pneumatic pressure test. The applied test pressure shall be not less than 1.3 times the specified external design pressure; or
- A vacuum test conducted at the lowest value of specified absolute internal design pressure. In conjunction with the vacuum test, a leak test shall be performed following a written procedure complying with the applicable technical requirements of Section V, Article 10 for the leak test method and technique specified by the user. Leak testing personnel shall be qualified and certified as required by Section V.

What is the minimum hold time of hydrostatic test pressure for pressure vessels?

ASME VIII-1 does not specify the minimum hold time for the hydrostatic test pressure. Normally it is determined by the quality procedure of the manufacturing shop or by AI. Thirty minutes to one hour is generally followed. When it comes to Shell and Tube heat exchangers, TEMA (Tubular exchanger Manufactures Association) specifies minimum test duration as 30 minutes.

How are pressure vessels inspected after the hydrostatic test?

Following the application of the hydrostatic test pressure, an inspection shall be made of all joints and connections. This inspection shall be made at a pressure not less than the test pressure divided by 1.3. Except for leakage that might occur at temporary test closures for those openings intended for welded connections, leakage is not allowed at the time of the required visual inspection. Leakage from temporary seals shall be directed away so as to avoid masking leaks from other joints.

The visual inspection of joints and connections for leaks at the test pressure divided by 1.3 may be waived, provided:

- A suitable gas leak test is applied;
- Substitution of the gas leak test is by agreement reached between Manufacturer and Inspector;
- All welded seams that will be hidden by assembly are given a visual examination for workmanship prior to assembly;
- The vessel will not contain a “lethal” substance.

Which medium may be used for hydrostatic test pressure of pressure vessels and what are the limitations regarding the pressure vessel temperature during the hydrotest?

Water is generally used, however any nonhazardous liquid at any temperature may be used for the hydrostatic test if below its boiling point. Combustible liquids having a flash point less than 110°F, such as petroleum distillates, may be used only for near atmospheric temperature tests.

It is recommended that the metal temperature during hydrostatic test be maintained at least 30°F above the minimum design metal temperature (MDMT), but need not exceed 120°F, to minimize the risk of brittle fracture. The test pressure shall not be applied until the vessel and its contents are at about the same temperature. If the test temperature exceeds 120°F, it is recommended that inspection of the vessel be delayed until the temperature is reduced to 120°F or less.

Can the pressure be painted before the hydrotest?

Unless permitted by the user or his designated agent, pressure-retaining welds of vessels shall not be painted or otherwise coated either internally or externally prior to the pressure test. When painting or coating prior to the hydrostatic test is permitted, or when internal linings are to be applied, the pressure-retaining welds shall first be leak tested in accordance with Section V, Article 10. Such a test may be waived with the approval of the user or his designated agent. Vessels for lethal service shall not be painted or otherwise coated or lined either internally or externally prior to the hydrostatic pressure test.

References:

ASME Boiler & Pressure Vessel Code, Section VIII, Division 1

SYSTEMATIC PROCEDURE FOR SELECTION OF HEAT EXCHANGERS

[*This article is reproduced from a paper (with same title as this article) published in "Proceedings of Institute of Mechanical Engineers, Volume 197A" in January 1983. The paper was addressed to the engineer responsible for preparing the heat transfer equipment datasheet, on which will be specified all the parameters required by the vendor for tendering for the equipment. Initially, the engineer must decide the type of equipment best suited for the application; guidelines for this selection are given in this Part A of the paper. Part B of the paper (not reproduced here) gives guidelines for preparing shell-and-tube heat exchanger datasheets.*]

This discussion of the selection of heat exchangers will, in general, be limited to those in the process industries, where a large number of different types are used in a great variety of services.

Heat transfer is the exchange of heat between two or more fluids, and may include condensation or evaporation of one or more fluids. The fluids themselves can be single compounds or mixtures. Temperature and pressure levels may significantly differ for different duties. Fluids may be clean, or may foul or scale excessively, or may be corrosive. Economics may favor wide or close temperature approaches between the fluids. These factors all play a part in heat exchanger selection.

This article assumes that process conditions have been optimized correctly, and that the heat duties, pressures and temperatures have all been specified; physical properties for the fluids are assumed known or calculable. Where phase changes occur, it is assumed that a heat release curve exists. At this stage, the engineer must decide the most suitable heat exchanger type.

The familiar shell and tube type is the most common because of its versatility. However, it may not always be the cheapest. It is also the most complex in terms of selection, having the largest number of construction variables of all exchanger types-because of this, discussion of the shell and tube type will be deferred till last.

AIR COOLED HEAT EXCHANGERS (ACHE)

Process cooling usually requires air or water or a combination of the two. Cooling by air is more likely to prove economical where water is expensive and/or in short supply, or if legislation does not permit heat rejection into water.

Figure 1: Air Cooled Heat Exchanger

The ACHE (Figure 1) draws or blows air over banks of finned tubes through which passes the fluid to be cooled. ACHE's suffer from larger sink temperature fluctuations between day and night or winter and summer than do water cooled systems. Temperature extremes can be covered by running small water cooled trim exchangers downstream of the ACHE in hot weather, or by the use of steam coils or re-circulation of the warmed outlet air to the inlet in cold weather.

Air cooled heat exchangers occupy a larger spatial area than simple once-through water coolers (this may not be the case if a cooling tower has to be used), but can be mounted well above grade, usually above pipe racks. If not allowed for in the design, fan-generated noise can be a nuisance to operators and nearby residents.

Air cooled heat exchangers can be designed for high pressures and temperatures, although, because of the search for efficient process conditions, temperatures seldom exceed ambient by more than 200-300^{o}F.

Design is usually carried out by specialists who will optimize such criteria as:

a) Capital versus running costs

b) Induced versus forced draught fans (induced = better air distribution and hailstone protection; forced = less power and easier maintenance)

c) Type and spacing of fins

d) Number of tube rows

e) Noise, type and design of fans

For such optimizations to be made, the process engineer will typically need to supply utility costs, ambient temperatures for winter and summer, height above sea level, payout period, noise level criteria, etc. He must also specify any special conditions such as fluid pour points or proximity of other heat sources such as furnaces or other air cooled heat exchangers. It is often necessary to obtain design and price for the complete air cooled heat exchanger system before a true comparison with the alternative water cooled system can be made. The cost of the alternative water cooled systems must also allow for their components. For water circulating through a cooling tower, this will include the cost of the cooling tower (not discussed here), associated pumps and piping, and operating costs, including makeup water, water treatment, power and maintenance. With a once-through system the costs must include the pumps, filters and piping and operating costs including water treatment, power and maintenance.

A proper evaluation of the alternative schemes for heat rejection depends on a number of factors including site location, local legislation, meteorological conditions, energy costs, payout period, etc. These factors can be evaluated by the user if he has adequate in-house data, but usually will require liaison with the vendor, particularly on prices.

<u>Process heat transfer</u>: All the remaining heat exchangers discussed in this article are designed for heat exchange between process fluids, although some may also be used specifically for heat rejection.

Note that the air cooled heat exchanger principle for increasing the surface area by finning on the side with the poor heat transfer coefficient also holds good in many other heat exchanger types. For example, low finned tubing is often used in shell and tube heat exchangers to improve the heat transfer effectiveness when, say, a gas flows on the shellside against a tubeside liquid.

PLATE HEAT EXCHANGERS (PHE)

Used in the food industry for many years, this type is now increasingly used in the process industries. It is displacing the shell and tube heat exchanger in some applications because it has superior ratios of heat transfer to cost, space and weight. More exotic the material, better the cost ratio is for PHEs. Weight and space are particularly expensive offshore.

A number of corrugated plates are clamped in a frame. The fluids between which heat transfer takes place enter through ports, gasketed so that each fluid flows up or down the alternate spaces formed by the plates (Figure 2). The periphery of all plates is also gasketed to prevent egress of fluids to the atmosphere. By suitable intermediate headers, two or more duties can be accommodated within a single frame. The corrugations on the plates give mechanical support and promote turbulence which enhances heat transfer and tends to reduce fouling. Transition to turbulence occurs at lower velocities than within shell and tube heat exchangers which makes the performance of the PHE superior in transferring heat to or from viscous fluids.

Close temperature approaches (i.e. thermally long duties) can be achieved because the flow is very nearly counter-current, and there is no possibility of flow bypassing heat transfer surface as there is with shell and tube heat exchangers. The PHE has a low hold-up volume and is especially suited to heat-sensitive fluids requiring a short residence time. Manufacturers claim that heat transfer coefficients are more than double those for the equivalent shell and tube heat exchanger. The PHE is less advantageous for heat exchange between fluids with significantly differing flowrates.

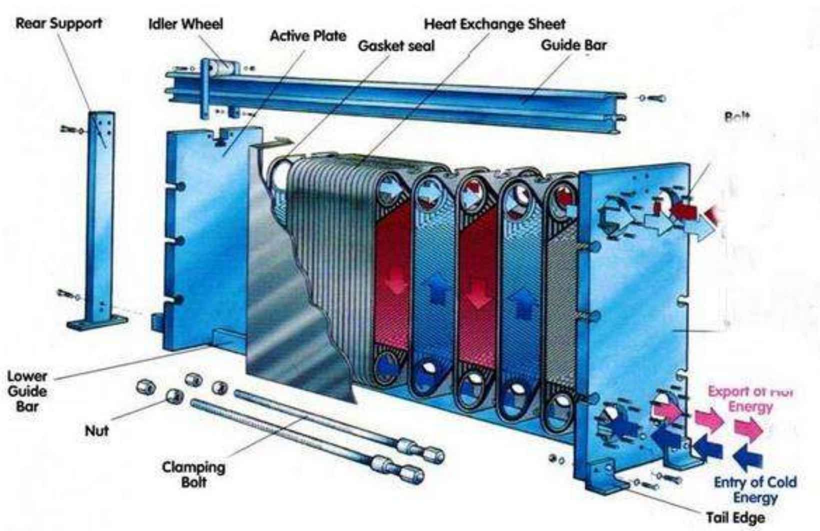


Figure 2: Plate Heat Exchanger

The plate exchanger is easy to clean since it is possible to remove all the plates individually. It is also versatile in that more plates can often be added to a given frame at a future date. The plates can be manufactured from any pressable material, e.g. stainless steels, titanium, Monel, cupro-nickel, etc. Gasket materials are the major limitation to PHE applications; manufacturers continue to attempt to extend their pressure and temperature ratings and solvent resistance. Currently, gaskets in elastomers are limited to 350 psi up to 200°F and in CAF to 160 psi up to 390°F. Special designs outside these ranges are sometimes possible. Gaseous or condensing fluids are hard to handle in PHEs because of the small port areas, and for similar reasons the PHE is unsuitable for solid suspensions of particles larger than 2 mm diameter. Surface areas range from a few square feet up to 12000 ft^2.

Rating can be carried out by following published data, but ultimately the PHE must be fabricated by a specialist vendor, and in order to obtain his guarantee he will insist on carrying out the rating himself. The major vendors do however allow free use of their thermal programs by their clients for initial design comparisons and economic evaluations (usually versus the equivalent shell and tube designs).

Mechanical design codes are more pertinent to the retaining frame; the plates and gaskets are however subject to manufacturers' works' tests. Maintenance and repair, as with cleaning, are especially easy with PHEs and require no special bundle pulling equipment nor removal of large numbers of bolts nor associated pipework. Gaskets may need replacing at regular intervals. For one pass PHEs, it is possible to locate all the pipe connections at one end of the frame which enables the unit to be opened without loosening any piping connections.

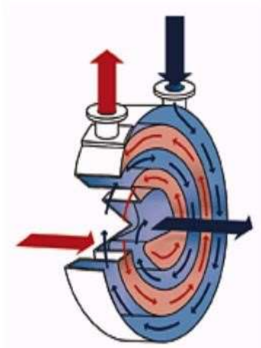
SPIRAL PLATE HEAT EXCHANGERS

This exchanger type is exceedingly compact (Figure 3). Two metal strips are rolled together in a spiral with studs to space them apart, rather like a Swiss roll. The strips are usually welded top and bottom sealed from each other. Gasketed cover plates are then bolted top and bottom to complete the channel seals. The SHE is made in three types. In one type, design is intended for counterflow in adjacent channels which enables the SHE to handle thermally long duties (large temperature crosses). One fluid enters at the center and exits at the periphery, the other vice versa.

Both channels can be cleaned mechanically by removing the appropriate cover plate. Vertically mounted SHE's are usually mounted on trunnions to gain easy access to both covers. The SHE is therefore well suited to fouling or scaling duties although it is not so easy to clean as a PHE.

The SHE is especially suited to slurries and liquids with suspended fibres and is therefore often employed in the pulp and paper industry and in mineral ore treatment where it can handle up to 50% w/w solids. The smooth channels of constant cross-section permit much higher velocities without erosion than do shell-and-tube heat exchangers. This factor is also of benefit in severe fouling duties where it is preferable to design for high velocities which tend to prevent fouling rather than to increase the surface as a safety margin for fouling which does the opposite. The possibility for accepting high velocities and the single channel on each side makes chemical cleaning in situ particularly effective.

The channel curvature causes secondary flow which creates turbulence at lower velocities than with the shell-and-tube heat exchanger; this makes the SHE more efficient at handling medium or high viscosity fluids. The SHE is the first choice for extremely high viscosities, say up to 500,000 cP, especially in cooling services, because maldistribution, and hence partial blockage by local overcooling, is less likely to occur in an essentially single channel exchanger. The lack of maldistribution and high permissible velocities make the SHE very suitable for processing heat sensitive fluids.



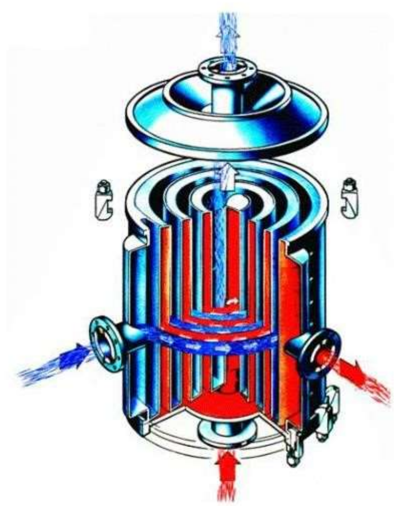


Figure 3: Spiral Plate Heat Exchanger

The surface area of each of the two spirals is of course approximately the same so when possible it is more economical to use this form of SHE for exchange between fluids having similar heat transfer coefficients. However, the plate spacing of each channel can be designed for a range of cross-sectional areas thus permitting a wide ratio of volume flowrates between the two fluids.

Both heat transfer surfaces can be augmented by corrugations, ribs, etc., and because both surfaces can be cleaned these pose little risk of unmanageable fouling.

The second type of SHE is designed for cross-flow operation. One channel is seal welded top and bottom. No channel plates are used; instead headers are installed top and bottom for two phase flow in cross-current with the coolant or heating medium in the spiral channel. It is particularly useful for ‘in-column’ re-boiler and condensing duties.

The third type is a variation of first type and allows for condensation in parallel across the majority of spiral turns with the outermost 2 or 3 turns in series for sub-cooling. It is particularly suited to ‘in-column’ condensation.

Design pressures depend on diameter but are typically around 200 psi. Design temperatures can be up to 750°F. The largest units are around 3000 ft^2. Materials can be any weldable material that can be cold formed, e.g. stainless steel, titanium Monel, etc. Maintenance is generally easy but repairs to damaged spirals are difficult.

The SHE is generally intermediate in price between the PHE and shell-and-tube heat exchanger for a given surface area but the SHE area can be significantly more efficient in heat transfer than that of the equivalent shell-and-tube heat exchanger. Design is usually done by the ‘vendor but can be carried out by use of published. Conventional mechanical design codes can be applied.

PLATE-FIN HEAT EXCHANGERS (PFHE)

Usually made in aluminum alloy, the PFHE has found extensive application in low temperature processing such as the cold section of air separation plants, hydrogen separation plants and natural gas liquefaction plants. There are also increasing applications at higher temperatures, e.g., on plants handling LPG, ethylene, enriched uranium or xylenes.

Normally, secondary surface is less efficient than primary surface because of fin inefficiency, so it does not pay to add secondary surface to both sides - it is cheaper to increase the primary surface instead. The exceptions are (a) where primary surface is expensive because it must be thick to retain the pressure, (b) space is at a premium, and (c) the heat fluxes are very low (i.e. high fin efficiency). The cryogenic application fits the last category and in addition, the secondary finned surface confers additional strength to the assembly. The PFHE (Figure 4) offers significant advantages wherever there are clean fluids with thermally long duties.

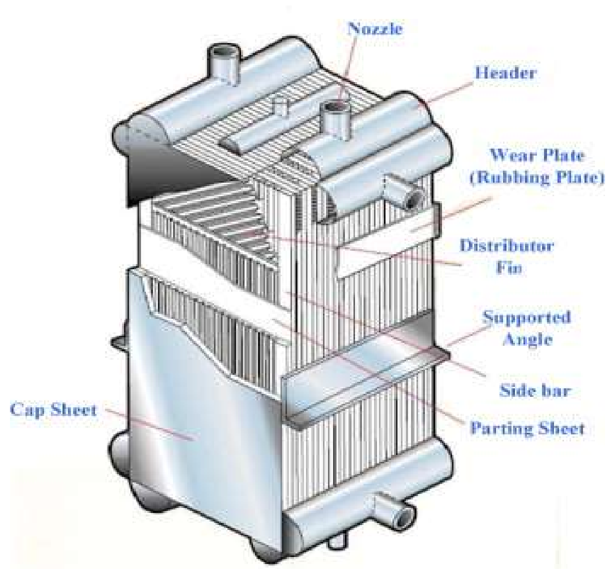


Figure 4: Plate-Fin Heat Exchanger

Layers of corrugated fins are dip-brazed between flat plates, in a sandwich fashion. Fluids in adjacent channels exchange heat through the finned surface and plates. Fin types can be varied to suit specific heat transfer and pressure drop requirements. Because the finned surface does not have to withstand a pressure differential, it can be very thin yet still be effective in heat transfer. Consequently the PFHE packs a very high surface area per unit weight and volume, and is cheap per unit heat load.

The method of construction prevents mechanical cleaning so restricts the PFHE to clean, solid free fluids, or to deposits which can be removed by solvent cleaning, back-flushing, or by warming up.

Most common applications for PFHEs are therefore in cryogenic services where their only competitor is the coiled-tube heat exchanger. The PFHE can handle more than ten streams and any of these may enter or exit part way along the exchanger. All streams flow over finned surfaces although the fin design may be specific to the process stream. The design may be made truly counter-current and can achieve very tight temperature approaches, as low as 2°F. Because of its short fin length, the PFHE is very suitable for heating or cooling viscous (but clean) fluids. Uniform flow distribution between the many parallel channels is very important where tight temperature approaches are required, so considerable care is necessary in the design of the inlet and outlet distribution channels.

Corrosion resistance is limited by the range of materials which can be formed and brazed successfully. PFHEs are usually made in aluminum alloy which can be brazed at relatively low. Other materials, such as stainless steels, nickel, copper, Inconel, titanium or alloys of them, can also be used, but brazing is always done with copper or nickel with which fluids must also be compatible; sizes are restricted by the dimensions of the furnace required to achieve the higher brazing temperatures.

Design pressure is limited by size but rises to about 2,300 psi for the smaller units. But if frequent pressure cycling is likely, the possibility of fatigue must be considered with materials, such as aluminum, which have no endurance limit stress. For example, 'switching' PFHEs in alloy 3003 are limited to 150 psi.

Maximum design temperatures are limited to about 150°F for aluminum alloys which can lose mechanical strength and become susceptible to corrosion at higher temperatures. For these reasons PFHEs for service temperatures around or above ambient are generally made from copper bearing alloys, and for these materials the maximum design temperature is limited by the brazing itself.

There are no restrictions on minimum service temperatures for the low temperature range of aluminum alloys. Thermal stresses can be high in cryogenic service especially at start-up, shut-down or during upsets, consequently a maximum process fluid bulk temperature differential is often set.

Surface areas in excess of 100,000 ft^2 are possible. Design of PFHEs is highly specialized and must be left to the vendor. It is worthwhile to obtain the vendor's opinion at the outset as to whether or not the application would suit a PFHE.

Mechanical design codes can be applied to PFHEs, and the finished units are generally works tested hydraulically to full test pressure on each stream independently.

COILED TUBE HEAT EXCHANGERS (CTHE)

The coiled tube heat exchanger, also known as Hampson heat exchanger competes with the plate fin heat exchanger in the cryogenic field, and can also be used for wider services. It can be constructed in a wider range of materials than the PFHE.

Small bore ductile tubes are wound in layers around a central mandrel or core. Each successive layer is wound the opposite hand to its predecessor from which it is separated by spacing strips. Tubes in individual, or groups of, layers may be brought together into one or more tube plates through which different fluids may be passed counter-current to the single shellside fluid. Plain or finned tubes can be used and overall surface areas can run up to 200,000 ft^2. Materials are usually aluminum alloys for cryogenics or stainless steels for higher temperatures which can be as high as 1350°F.

Because the small passages on both sides of the exchangers do not permit mechanical cleaning these exchangers are used for clean, solid free, fluids only or for fluids whose deposits can be removed by back-flushing or by periodic washes with solvent. The exchanger is very compact but slightly less so than a plate fin heat exchanger; however, several of the PFHE disadvantages are overcome by the CTHE. Differential thermal expansion is less of a problem, dip brazing is not required and design pressures and temperatures can be greater than in PFHEs. Close temperature approaches are also possible when the CTHE is operated in counter-flow. The CTHE is also useful for heat exchange to viscous fluids because of its low L/D ratio, a function of the tube's continuous helix.

The design of a CTHE can be exactly matched to process requirements. The spacer thickness on the shellside can be adjusted to take up all the allowable shellside pressure drop and tube bores and lengths can be adjusted to absorb all the allowable tubeside pressure drop. Even distribution into the shell can be achieved by use of a thin metal shroud. The parallel paths on the shellside are all the same length as they are on the tubeside, which also ensures even distribution. Hence maximum effect temperature driving force occurs throughout the length of the exchanger.

The CTHE is not cheap because of the material costs, high labor input in winding the tubes, and because the central mandrel is useless for heat transfer but increases the shell diameter. In practice shell diameters tend to be standardized.

DOUBLE PIPE HEAT EXCHANGERS (DP)

Double pipe heat exchangers are suitable for high pressures and temperatures, and for thermally long duties. Simple in construction, their usual application is for small duties requiring, typically, less than 300 ft^2 surface. They are not very compact.

As implied by their name, one pipe is mounted concentrically inside another, and heat exchange is pure counter-flow. The units can be designed to be taken apart for cleaning, but more often they are simply all welded jacketed pipes, the annulus of which cannot be mechanically cleaned. Units can be connected in serpentine fashion but their cost rapidly escalates until a small shell-and-tube heat exchanger becomes relatively cheaper.

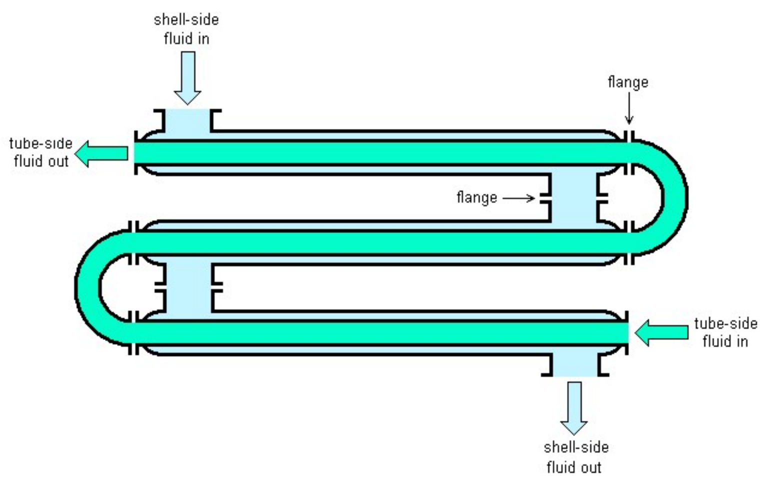


Figure 5: Double Pipe Heat Exchanger

Design can easily be done by hand using published references that cover the annulus heat transfer coefficient. Alternatively simple design programs can be written or purchased. Proprietary designs using double pipes with longitudinal fins within the annulus, are used where additional surface is required relative to the inner pipe – for example, where steam is employed in the inner pipe to heat viscous oil in the annulus prior to entering a pump. Design is usually left to the vendor but is reasonably straightforward provided the fin efficiency is allowed for.

Design pressures and temperatures depend on the method of construction but are broadly similar to shell-and-tube heat exchangers.

GRAPHITE HEAT EXCHANGERS (GHE)

Carbon, in the form of impervious graphite, has a high thermal conductivity relative to most other heat exchanger materials. Graphite also has a high resistance to corrosion, being attacked by only a few chemicals, typically strong oxidizing agents such as aqua regia or chromic acid. Provided stresses are kept compressive, graphite can be used successfully as a heat exchanger material, of special use when dealing with highly corrosive fluids. In some services, typically hydrogen chloride absorbers or sulfuric acid dilution coolers, there are few other suitable and economical materials.

Graphite is usually made from finely ground coke bound with plastic hydrocarbons. This material is pressure-molded to the desired form and then pyrolysed, first to decompose and drive off the binding materials and then to polymerize the carbon to the hexagonal crystal lattice form of graphite at about 5000°F. The resulting graphite is impregnated with resins to make it impervious.

Graphite is immune to thermal shock and can be used in liquid service up to 400°F or up to 1500°F in some gas services, usually with cooling water on the other side. Graphite has no metallic ions to contaminate fluids so finds special application in the food and drug industries.

Because graphite is soft and easily machinable, a wide variety of heat exchanger types can be made. The most common type is the graphite cubic block (Figure 6). Bolted plates ensure the graphite block is held in compression. The two (or sometimes more) process streams pass through holes or slots drilled at right angles to each other. Slots are designed to double the surface area on one side for applications such as cooling vapors.

Figure 6: Graphite Block Heat Exchanger (*Cepic*)

The cubic block is ideal for corrosive fluids on both sides. Pass arrangements can be made to suit individual stream requirements. Surface/unit ranges from 2 to 300 ft^2 and pressures up to 150 psi.

Another common type is the modular block which consists of machined cylindrical graphite modules compressed end to end into a steel shell by external tie rods. The process fluid passes axially and the service fluid radially, through drilled holes. This design is commonly used for evaporators, re-boilers, falling film absorbers (e.g. for HCl) and for hot gas cooling (up to 1500°F). Surface/unit ranges from 1 to 2000 ft^2; process pressures are limited to 85 psi.

The passages of both the cubic and modular block designs have relatively low length/diameter ratios; these are advantageous for handling highly viscous flows whose heat transfer coefficients would otherwise be low because of laminar flow. Fouling and scaling tendencies are reputed to be low, and chemical cleaning is very applicable because of graphite's resistance to corrosion.

Shell and tube exchangers can also be built with tubes, tubesheets and channels in graphite and the remainder generally from stainless steel and Tufnol. Surfaces are 200 ft^2/unit upwards but tubeside pressure is limited to 44 psi.

Ratings can be carried out by the user fairly straightforwardly provided the manufacturer's standard dimensions are known. The shell and tube design is rated as such. However, graphite exchangers must be priced by the vendor and the ratings can also be left to him.

SCAPED SURFACE HEAT EXCHANGERS

The most common scraped surface exchangers are essentially double pipes using a helical screw within the inner pipe to extract the solids precipitated from the fluid being cooled. Chief applications are in de-waxing plants and the food industries.

For a totally different application - the evaporation from, or heating of, heat sensitive fluids-the high speed wiped film evaporator can be used. A very thin fluid layer is created by centrifuging from a rotating bladed impeller with close tolerance to the inner wall of a heated cylindrical surface. The residence time is thus very short (Fig. 13).

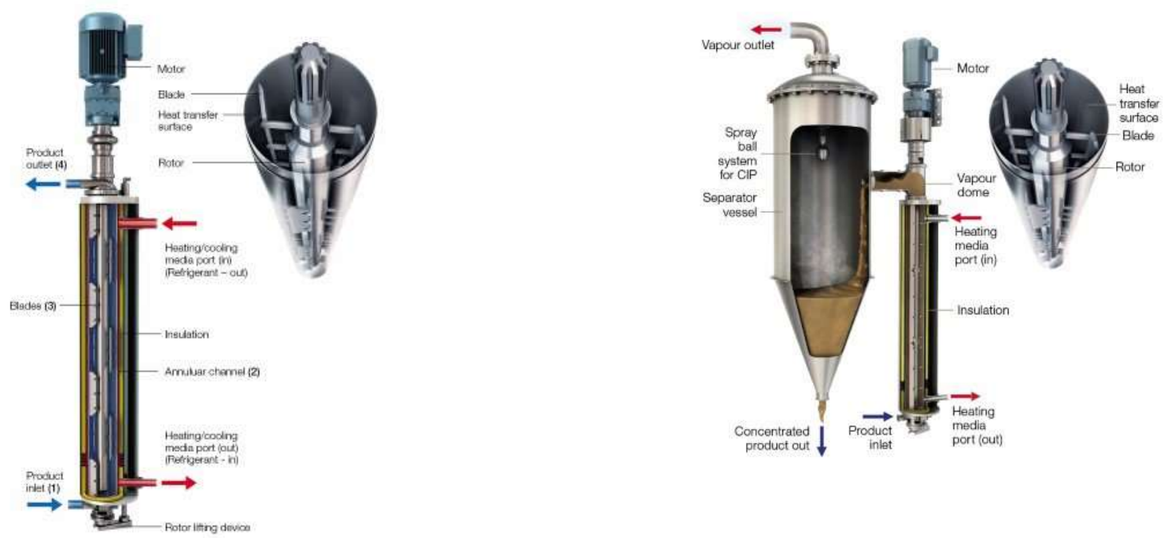


Figure 7: Scraped Surface Heat Exchanger (*Alfa Laval*)

For both types of scraped surface heat exchangers, operating and capital costs are high, and their applications are highly specific. Design should be left to the vendor.

SHELL-AND-TUBE HEAT EXCHANGERS

This type remains dominant in the petrochemical/ offshore industries. Figure 8 shows a typical design. Relatively small tubes, ½ - 1 in. OD, are expanded or welded into tubesheets to make a bundle which itself is enclosed within a shell. Tube lengths typically range from 8 to 24 feet for many applications although even longer tubes are sometimes used. The fluid passing through the tubes exchanges heat with that passing over the tubes on the shellside. Tube-to-tubesheet welding is used when it is imperative no leakage occurs between shell and tubesides or when repeated stressing may slacken the cheaper expanded joint.

The number of tubeside passes can be varied, and more usually is an even number. The shellside usually has transverse baffles to direct the shellside fluid from one side to the other down its length, thereby promoting heat transfer.

Thus the usual shell and tube exchanger has one shell pass and two or more tube passes. This configuration leads to a mixture of co-current and counter-current flow, which will reduce the effectiveness of the temperature driving force; a correction factor has therefore to be applied. In duties where the exit temperatures are similar, or even overlap, it may prove necessary to split the duty between two or more such shells in series. Alternatively, a single tube pass may be used to ensure counter-current exchange takes place. The cheapest form of single pass exchanger uses two fixed tubesheets. The bundle cannot be withdrawn and mechanical cleaning of the outside of the tubes is not possible. Furthermore, if the temperature differential between the shell and tube material causes too great a stress, some form of stress-relieving device such as a bellows must be used. If either the pressure or temperature falls outside the design range of the stress-relieving device, then a single tubeside pass is not feasible. Two passes on each side using a floating head or U tube may be considered; the shell containing a longitudinal baffle to direct its flow counter-current to the tubeside flow.

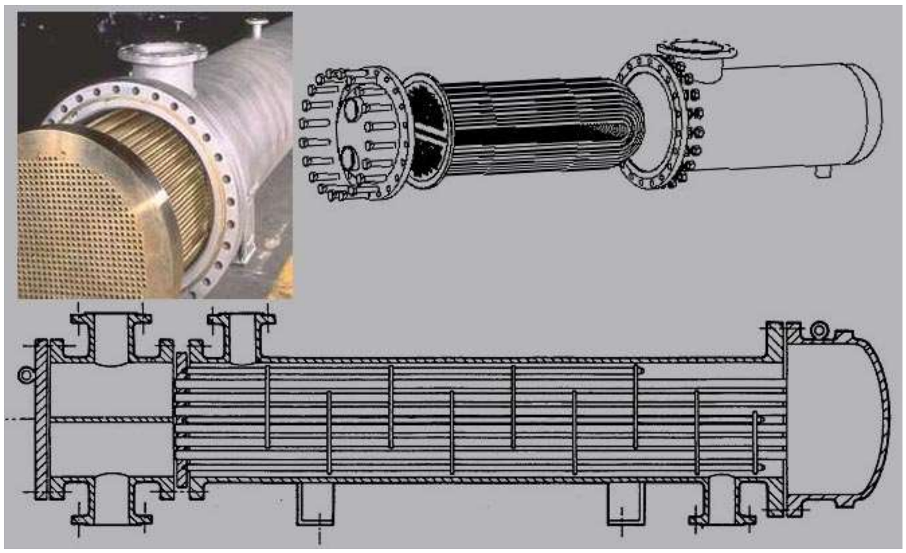


Figure 8: Shell-and-Tube Heat Exchanger (*SEC Heat Exchangers*)

If the bundle does not have to be removed, this longitudinal baffle should be welded to the shell wall to prevent a short-circuit leak between baffle and shell at the maximum pressure differential, i.e., the inlet to outlet. If the bundle has to be removed a sprung seal can be used on the longitudinal baffle edge, but this is still prone to leakage, especially after the bundle has been pulled once or twice. An alternative design uses a dispensable light gauge shroud to which the longitudinal baffle is welded.

Because of the necessary fabrication tolerances, not all the shellside fluid crosses the tubes in the most efficient manner; indeed, leakage between the bundle and the shell results in almost no heat transfer to the leakage stream which can be very serious in duties such as viscous fluid cooling, because the hot bypassing fluid sees a far lower pressure loss than the stream cooled within the bundle. Thermal designs have to make allowance for these inefficient streams and, because they all act simultaneously in parallel, a computer program is usually required to find a design. There are several such programs available, the most common and best authenticated being HTRI. Whatever program is used, however, it is necessary for the user to be something of an shell-and-tube design specialist to obtain the best results, and consequently thermal designs are usually left to the fabricator or to an shell-and-tube design consultant.

Shell and tube heat exchangers may be made in almost any combination of materials, although other exchanger types in the same materials may often be cheaper. Sometimes low-finned tubing is used to extend the external surface area. The fins are rolled into tubing usually two gauges thicker than plain tubes so that the final outer diameter is the same or just less than the original plain tube outer diameter. Low-finned tubing can thus be threaded through the bundle cage on the same pitch as for plain tubes. Finned tubes are likely to be economical wherever the outside heat transfer coefficient is significantly poorer than the inside coefficient, where space and weight are particularly expensive (e.g. offshore oil and gas production), or to increase the possibilities of holding internal fluid velocities within a desired narrow range (e.g. glycol injection services).

The usual mechanical design codes, ASME, BS, etc. can all be applied to shell-and-tube heat exchangers. Maintenance is straightforward but can be quite tedious; for example, the replacement of a gasket on a split ring floating head exchanger will require unbolting two girth flanges and removing two dished ends before the gasket can be removed and replaced. Heavy duty tackle or purpose built devices are required to pull bundles, especially if they are heavily fouled or corroded.

SELECTION OF EQUIPMENT TYPE

Table 1 summarizes the features of the heat exchangers discussed in this article. For a given application, there may only be one suitable equipment type; for example, if crystallization is intended, a scraped surface heat exchanger is indicated (note however that there are other forms of crystallizers not discussed here such as the external drum crystallizer). For many applications however, viable alternatives do exist and, for these, vendors should be asked to tender. Only then can a final selection be made. For example, flat square plate heat exchangers with flows at right angles are becoming popular for clean gas to gas heat exchange with low differential pressures in heat recovery systems. For dirty gas to gas heat exchange, glass tubular exchangers are being used. These have square tube bundles with modules of tubes a few rows deep which can be pulled out like a drawer for cleaning. Heat pipe heat exchangers are also used for clean gas/gas services.

Table 1: Heat Exchanger Selection Criteria

Criteria	Exchanger Type								
	ACHE	PHE	SHE	PFHE	CTHE	DP	GHE	SSHE	S&T
Pressure, psi	6,000	300 [4]	250 [7]	1,000 150 [11]	1,000	600	150	600 [18]	8,000
Temperature, °F	[1]	500 [4]	750	150 [12] 750 [13]	900	1,000	[16]	600	1,000
Max ft^2/ Unit	None	16,000/ Frame	3,000	100,000 [12] 500 [13]	200,000	300 [14]	[17]	10	30,000/ Shell
Compactness	x [2]	xxxx	xxxx	xxxxx	xxxx	x	xxx	x	x
Mechanical Cleaning	xx	xxxxx	xxxx	x	x	xxx	x	xxx	xxx
Chemical Cleaning	xx	xxxx	xxxx	xx	xxx	xxx	xxxxx	xxxx	xxx
Cost/ ft^2	xx	xxxx [5]	xxx [8]	xxxxx	xxxx	xx	x	x	xx
Maintenance Ease	xx	xxxxx	xxxx	x	x	xxx	x	xx	xx
Corrosion Risk	xxx	xxxx	xxxx	xxx	xxxx	xxxx	xxxxx	xxx	xx
Fouling Risk	xx	xxxxx	xxxx	xx	xxx	xxx	xxx	xxxxx [18]	x
Fouling Effect	x [3]	xxxx	xxxx	x	xx	xxx	xx	xxxx [18]	xx
Leakage Risk	xx	x [6]	x [9]	xxxx	xxx	xxx [15]	x	xx	xx [22]
Duty Changes after Installation	xx	xxxx	x	xxx	xx	x	xxx	x	x
Temperature Cross	x	xxxx	xxxx	xxxxx	xxxxx	xxx	xxx	xxx	xxx
Viscous Flow	x	xxxx	xxxx	xxxxx	xx	xx	xxxx	xxx	x xxx [23]
Heat Sensitive Fluids	xx	xxxx	xxxx	xxx	xx	xx	xxx	xxxxx [19]	x
Solids Flowing	x	xx	xxxx	x	x	xxx	x	xxxxx [20]	x
Gases	xxxx	x	xxx	xxxx	xxxx	xxxx	xxx	x	xxxx
Phase Change	xxxx	x	xxxx	xxxx	xxxx	xxxx	xxx	xxxxx [21]	xxxx
Multi Fluid Exchange	xxx	xxx	x	xxxxx	xxxx	x	xxx	x	xx

x - Very poor, xx - Poor, xxx - Fair, xxxx - Good, xxxxx - Very Good

1. Use restricted to temperatures from ambient to around 300ºF.
2. Often mounted high up or above pipe racks
3. Fouling on outside can reduce air flow and diminish MTD.
4. Depends on gasket material.
5. Low relative cost applies to non-ferrous materials.
6. Plate edges can be seal welded but dismantling then very difficult.
7. For diameters up to 1 m. For larger diameter, the pressure limit is lower.
8. In all metals.
9. See note 6.
10. Available only in non-ferrous materials.
11. Applies to reversing service in aluminum.
12. In aluminum.
13. In non-ferrous, non-aluminum materials.
14. Above 300 ft^2, shell and tube exchangers are usually cheaper.
15. If all welded.
16. 400ºF for liquids, up to 1500ºF for some gases.
17. 300 ft^2 for cubic block; 2000 ft^2 for modular block.
18. Applies to scraped inside.
19. High speed rotor.
20. Low speed rotor.
21. Liquid to solid, or liquid to vapor.
22. Depends on TEMA type.
23. Applies to viscous fluids being heated on shellside.

It is always worthwhile carrying out an approximate rating using ball-park coefficients. The resulting required surface area will indicate whether a simple double pipe is economical (surface less than 300 ft^2), whether the area falls within the range of proprietary equipment such as plate heat exchangers, or whether a large number of shell and tube units will be required. Knowledge of this sort at an early stage for all the process duties will highlight those which are most worthwhile to study in more detail. For borderline cases, both in terms of range of application and in cost, it sometimes pays to continue to run with an alternative equipment type right up to order placement, although the overall project cost and schedule may dictate otherwise.

References:

Systematic Procedure for Selection of Heat Exchangers – A. Larowski and M.A. Taylor

CAN YOU REALLY MULTITASK?

Think you are good at doing several things at once?

Reading and listening to music? Driving and talking on the phone, or texting while sitting in a meeting?

THINK AGAIN.

To be sure, we multitask all the time. And there are several tasks that we can multitask really well. The bodily functions – blood circulation, breathing and other such functions – are designed to be performed simultaneously. However, here in this article, we are referring to multitasking functions that involve decision-making through the brain. Everything that we do (sans some critical bodily functions) has its origins in our brain. Using computer analogy, our brain is simply not capable of parallel processing. It is only capable of processing one task at a time; i.e. it is a serial processor. However, it switches between many tasks so rapidly that it seems they are being performed simultaneously – like reading and listening to music, driving and talking on the phone, or texting while sitting in a meeting.

First, let us define "multitasking":

- It can mean performing two or more tasks simultaneously.
- It can also involve switching back and forth from one thing to another.
- Multitasking can also involve performing a number of tasks in rapid succession.

Multitasking seems like a great way to get a lot done at once. In the past, many people believed that multitasking was a good way to increase productivity. While it may seem like you are accomplishing many things at once, research has shown that our brains are not nearly as good at handling multiple tasks as we like to think we are. In fact, some researchers suggest that multitasking can actually reduce productivity by as much as 40%. What you think of as multitasking is actually quickly shifting attention and focus from one thing to the next. Switching from one task to another makes it difficult to tune out distractions and can cause mental blocks that can slow you down. Multitaskers have more trouble tuning out distractions than people who focus on one task at a time.

<u>IMPACT OF MULTITASKING</u>

In order to determine the impact of multitasking, psychologists asked study participants to switch tasks and then measured how much time was lost by switching. The study conducted by Robert Rogers and Stephen Monsell found that participants were slower when they had to switch between tasks than when they repeated the same task. Another study conducted in 2001 by Joshua Rubinstein, Jeffrey Evans and David Meyer found that participants lost significant amounts of time as they switched between multiple tasks and lost even more time as the tasks became increasingly complex.

Multitasking is all too common. At any given moment, you might be texting a friend, switching between multiple windows on your computer, listening to the blare of television, and talking to a friend on phone – all at once! When we do get a quiet moment when nothing is demanding our attention, we might find ourselves unable to avoid the distraction of our favorite apps or social media sites.

<u>BUT IS MULTITASKING BAD FOR YOU</u>?

So while we know that multitasking is not good for your productivity, is it possible that it might actually be bad for your brain health? What impact does such a constant barrage of stimulation have on developing minds?

In one 2009 study, Stanford University researcher Clifford Nass found that people who were considered heavy multitaskers were actually worse at sorting out relevant information from irrelevant details. They were also much less mentally organized. What was surprising was that even when these multitaskers were focusing on just one single task, their brains were less effective and efficient; i.e. their cognitive processes were impaired. It was also found that there is correlation between the two – that is, negative impact on performance while performing a single task is a direct result of heavy multitasking.

The negative impact of chronic, heavy multitasking might be most detrimental to the adolescent mind. At this age in particular, teen brains are busy forming important neural connections. Spreading attention so thin and constantly being distracted by different streams of information might have a serious, long-term, negative impact on how these connections form.

SO WHAT SHOULD BE DONE TO AVOID THE POSSIBLE DELETERIOUS IMPACT OF MULTITASKING?

It is recommended that you follow the guidelines given below:

1. Limit the number of things you juggle at any given time to just two tasks.
2. Instead of constantly switching back and forth from one task to another, try to devote your attention to one task for a 20-minute period before switching to the next task.
3. Certain situations demand that we fully refrain from multitasking – driving for example. Your attention should be 100% on the road while driving; do not text, do not talk on phone, or otherwise consult phone.

Brain doesn't really do tasks simultaneously; we just switch tasks quickly. Each time we move from hearing music to writing text or talking to someone, there is a stop/start process that goes on in the brain. That stop/start process is rough on us; rather than saving time, it costs time, it is less efficient, we make more mistakes, and over time it can be energy sapping.

References:

How multitasking affects productivity and brain health – Kendra Cherry

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