Cold-Plus™ for Reduced Drag Reduction and Increased Heat Transfer

By

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1.0 Abstract

This report is written to provide insight into why the addition of the Cold-Plus™ PTFE polymer reduces drag in refrigeration systems and increases the heat transfer to the working fluid. The friction drag is directly related to the flow Reynolds Number, and as the Reynolds Number increases with the addition of the PTFE additive, the resulting flow friction, f, is reduced as shown in the Moody Diagram (Figure 5). This reduction in flow friction results because the polymer interacts with the turbulent eddies and the viscous sublayer near the wall in boundary layer in the pipes (Figure 3).

For turbulent flow in a smooth pipe, a 15% increase in the Reynolds Number of the refrigerant with polymer additive results in approximately a 25% reduction in flow friction. As a result of this reduction in friction and drag, the system power requirements are reduced and there is an overall improvement in the system efficiency.

Also, because of the turbulent mixing of the fluid at the pipe wall, the fluid temperature will be increased. As the Reynolds Number of the flowing fluid is increased, the Nusselt Number will increase (Figure 7) and results in an increase in the heat transfer from the wall to the fluid. A 14% change in Reynolds Number will result in approximately a 25% increase in Nusselt Number and heat transfer. This increase in heat transfer will increase the performance of the A/C system.

The PTFE and the Cold-Plus™ particles are combined with other fluids to create a liquid treatment that is added to the circulating fluid of an operating AC system. Due to its lubricating properties, thermal conduction properties and propensity to create boiling sites on the inside surface of the evaporator coil, the particles increase the efficiency, including heat transfer efficiency, of the air conditioning system, thereby reducing its power consumption (the energy required to operate the compressor). Tests have indicated a 14% to 17% system energy

reduction. The beneficial operating features are realized within minutes of adding the particles to the fluid. The particles may also extend the life of system components and reduce the requirements for regular maintenance.

2.0 Introduction

It is known that a refrigerant typically exhibits substantial lubricating properties. Removing oil fouling and the resultant improvement in heat transfer in the evaporator are other aspects of improving efficiency in A/C and refrigeration units. Cold-Plus™ is a patent pending polymer that permanently bonds to all metal surfaces within the system and provides benefits other than the removal of oil fouling.

The addition of the polymer allows the flow of the refrigerant to occur with less flow friction resistance which results in an improved performance of the evaporator coils and reduced power consumption by the unit.

Additional benefits of friction reduction occur throughout the system. The molecular structure of the polymer provides friction reduction within the compressor, particularly within reciprocating compressors. This friction reduction shows in the reduced power for the compressor during the first minute of operation and throughout the life of the system.

The reduction in flow friction results because of the increased Reynolds Number (Re) that occurs with the addition of the additive to the refrigerant. As shown in the Moody Diagram (Figure 5) the friction factor, **f ,** decreases with increased Reynolds Number for both laminar and turbulent pipe flow. For turbulent flow in a smooth pipe, a 15% increase in the Reynolds Number of the refrigerant with polymer additive results in approximately a 25% reduction in flow friction. This decrease in friction results because of the interaction of the polymer with the turbulent boundary layer and viscous sublayer of the flow of refrigerant. The addition of polymer solution in the flow of refrigerant dampens the turbulence next to the wall of the pipe and reduces the flow friction and reduces drag.

The reduction in drag as a result of an increase in Reynolds Number is significant for lower, laminar flows. These types of flows occur during startup and with the

addition of the Cold-Plus™ additive, the friction drops significantly as shown in Figure 5. This will again provide significant reduction in the power required to start and operate the system.

 The addition of polymers in the refrigerant will reduce the flow friction to a maximum limit identified as the `MDR asymptote' as shown in Figure 6. Increasing the polymer concentration does not lead to drag reductions in excess of this asymptote (which is insensitive to polymer species, molecular weight, or the type of polymer). However, Figure 5 and Figure 6 both show that by increasing the flow Reynolds number with the addition of a polymer, the flow friction is reduced.

Also, as shown in Figure 7, the Nusselt Number increases with Reynolds Number for flow in a pipe. Since the Nusselt number is a function of the convection heat transfer coefficient, the increase in Reynolds Number will result in an increase in the heat transfer from the wall of the pipe to the working fluid. A 14% change in Reynolds Number will result in approximately a 25% increase in Nusselt Number and heat transfer.

This paper describes the effects of the interaction of the Cold-Plus™ polymer with the refrigerant boundary layer flow and how this turbulent mixing reduces the flow friction and increases the heat transfer to the fluid which results in an increase in the operating efficiency and life of the system.

3.0 Drag Reduction and Increased Longevity with Polymer Solutions

The flow of refrigerant through the compressor tends to remove a lubricant from the compressor surfaces that it contacts. The lubricant remains dispersed in the refrigerant, but when the lubricant reaches the relatively low temperature evaporator its viscosity increases. The lubricant thus tends to accumulate in the evaporator and starve the compressor of lubricant. If the amount of lubricant returning to the compressor is insufficient, the compressor lubrication cannot be maintained relatively constant over time. This condition negatively affects the compressor's operating life, raising the compressor frictional forces and requiring the motor to supply additional power to drive the compressor.

Cold-Plus™ and its PTFE boundary lubrication helps to eliminate this problem.

The boundary lubrication is caused by the unique properties of the polymers composed of PTFE particles of various sizes that are designed to permanently bond to metal contact surface. This bonding is maintained for the life of the system and provides for long term efficiency and reduced maintenance.

3.1 PTFE

PTFE is a fluorocarbon solid and is a high molecular weight compound consisting wholly of carbon and fluorine. Fluorocarbons are not as susceptible to the London dispersion forces (van der Waals force) due to the high electronegativity of fluorine. Therefore, water and water-containing substances, and oil and oil-containing substances, do not wet PTFE as adhesion to PTFE surfaces is inhibited. PTFE has one of the lowest coefficients of friction against any solid (0.05-0.10). Due to this property, PTFE is used as a non-stick coating for pans and other cookware. It is very non-reactive, partly because of the strength of carbon–fluorine bonds, and so it is often used in containers and pipework for reactive and corrosive chemicals. Where used as a lubricant, PTFE reduces friction, wear and energy consumption of machinery.

4.0 Boundary Layers in Pipes

When a fluid flows over a stationary surface, e.g. the wall of a pipe, the fluid touching the surface is brought to rest by the shear stress τ at the wall. The velocity increases from the wall to a maxi mum in the main stream of the flow as shown in Figure 1.
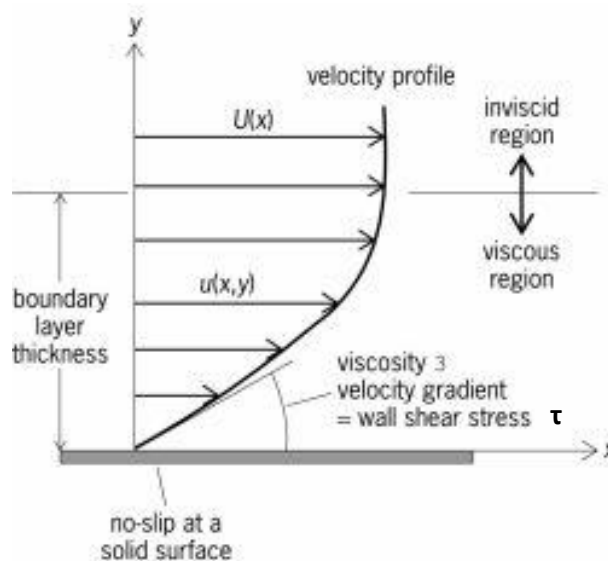

Figure 1 Boundary Layer Profile

Looking at this two-dimensionally in Figure 1, we get the velocity profile from the wall to the center of the flow. This profile doesn't just exist, it must build up gradually from the point where the fluid starts to flow past the surface - e.g. when it enters a pipe. If we consider a flat plate in the middle of a fluid as shown in Figure 2, we will look at the build- up of the velocity profile as the fluid moves over the plate. Upstream the velocity profile is uniform, (free stream flow) and a long way downstream we have the velocity profile shown in Figure 1. This is the known as fully developed flow.

This region, where the velocity profile in the flow due to the shear stress at the wall, we call the boundary layer. The stages of the formation of the boundary layer are shown in the Figure 2 below:
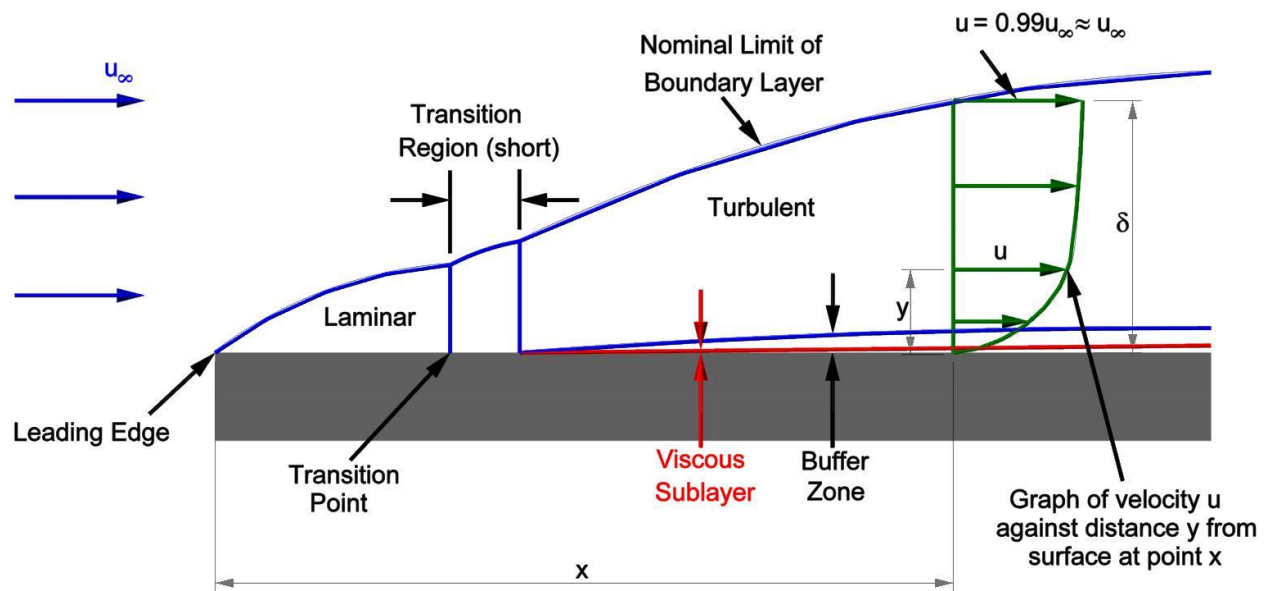

Figure 2 Boundary Layer on a Surface

We define the thickness of this boundary layer as the distance from the wall to the point where the velocity is 99% of the "free stream" velocity, the velocity in the middle of the pipe. The boundary layer thickness, δ, is the distance from wall to a point where $u = 0.99 \, u_{mainstream}$

The value of δ will increase with distance from the point where the fluid first starts to pass over the boundary. It increases to a maximum in fully developed flow.

Correspondingly, the drag force, D, on the fluid due to shear stress τ at the wall increases from zero at the start of the plate to a maximum in the fully developed flow region where it remains constant. Our interest in the boundary layer is that its presence greatly affects the flow through a pipe. Next we will examine some of the phenomena associated with the boundary layer and discuss why these occur.

4.1 Formation of the boundary layer

Above we noted that the boundary layer grows from zero when a fluid starts to flow over a solid surface. As is passes over a greater length more fluid is slowed by friction between the fluid layers close to the boundary. Hence the thickness of the slower layer increases.

The fluid near the top of the boundary layer is dragging the fluid nearer to the solid surface along. The mechanism for this dragging may be one of two types:

The first type occurs when the normal viscous forces (the forces which hold the fluid together) are large enough to exert drag effects on the slower moving fluid close to the solid boundary. If the boundary layer is thin then the velocity gradient normal to the surface, *(du/dy)*, is large so by Newton's law of viscosity the shear stress, $\tau = m\ (du/dy)$, is also large. The corresponding force may then be large enough to exert drag on the fluid close to the surface.

As the boundary layer thickness becomes greater, so the velocity gradient becomes smaller and the shear stress decreases until it is no longer enough to drag the slow fluid near the surface along. If this viscous force was the only action that occured then the fluid would come to a rest.

It does not come to rest but the second mechanism comes into play. Up to this point the flow has been laminar and Newton's law of viscosity has applied. This part of the boundary layer is known as the laminar boundary layer

The viscous shear stresses have held the fluid particles in a constant motion within layers. They become small as the boundary layer increases in thickness and the velocity gradient gets smaller. Eventually they are no longer able to hold the flow in layers and the fluid starts to rotate as shown in Figure 3.
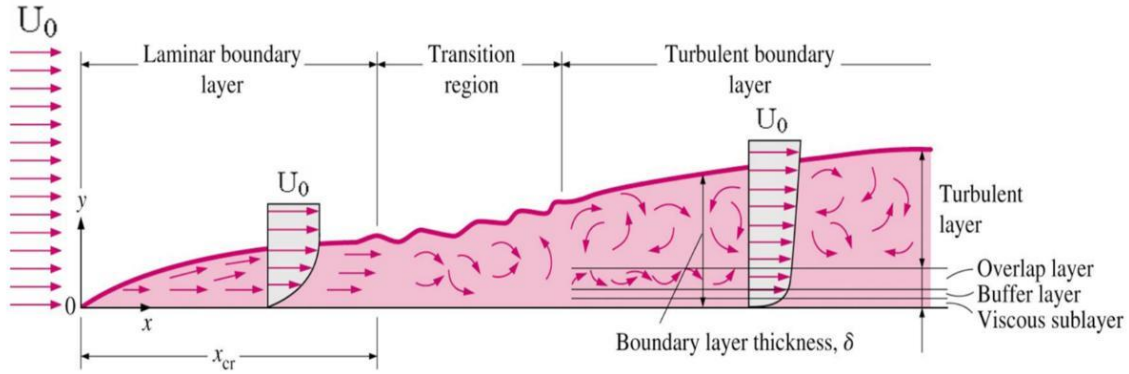
Figure 3 – Boundary Layer Growth

This causes the fluid motion to rapidly become turbulent. Fluid from the fast moving region moves to the slower zone transferring momentum and thus maintaining the fluid by the wall in motion. Conversely, slow moving fluid moves to the faster moving region slowing it down. The net effect is an increase in momentum in the boundary layer. We call the part of the boundary layer the turbulent boundary layer.

At points very close to the boundary in the viscous sublayer, the velocity gradients become very large and with the viscous shear forces again becoming large enough to maintain the fluid in laminar motion. This region is known as the laminar sub-layer. This layer occurs within the turbulent zone and is next to the wall and very thin - a few hundredths of a mm.

4.2 Boundary Layers in Pipes

As flow enters a pipe the boundary layer will initially be of the laminar form. This will change depending on the ratio of inertial and viscous forces; i.e. whether we have laminar (viscous forces high) or turbulent flow (inertial forces high).
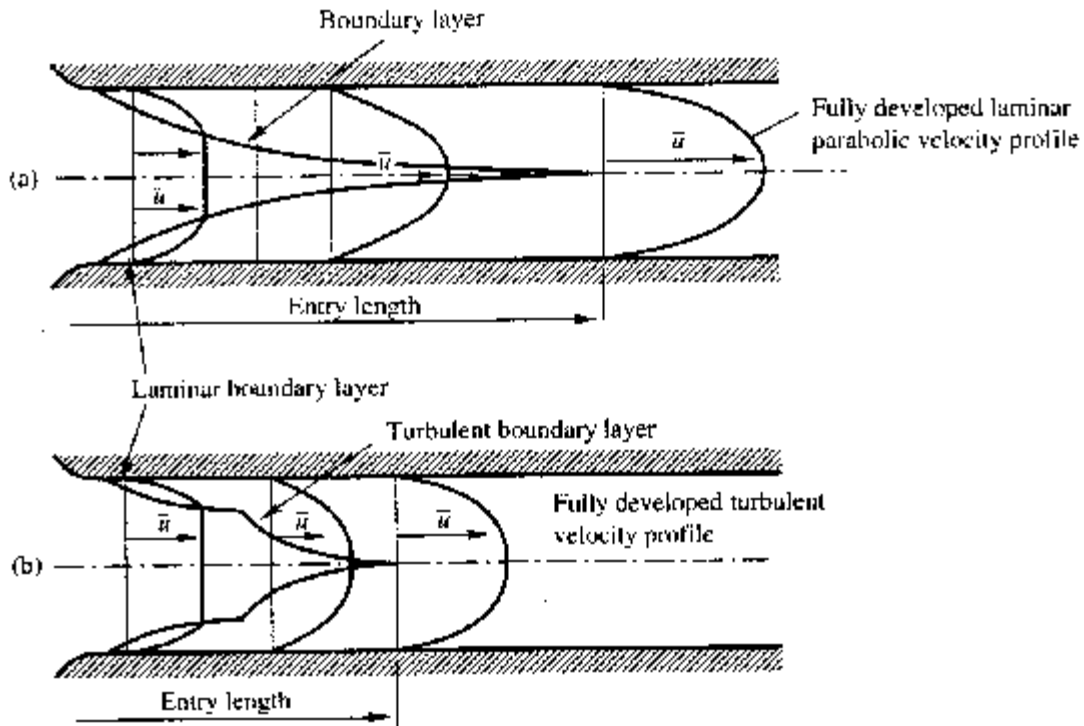
Figure 4 – Boundary Layers in Pipe Flow

To determine whether a particular flow in a pipe is laminar or turbulent the Reynolds number in Eqn. (1) is used:

$$Re = \frac{\rho u d}{\mu} \qquad (1)$$

(ρ = density u = velocity μ = viscosity d = pipe diameter)

Where,

Laminar flow: $Re < 2000$,
Transitional flow: $2000 < Re < 4000$,
Turbulent flow: $Re > 4000$

If we only have laminar flow the profile is parabolic as only the first part of the boundary layer growth diagram is used. So we get the top diagram in the above Figure 4a.

If the flow is turbulent (or transitional), both the laminar and the turbulent (transitional) zones of the boundary layer growth diagram are used. The growth of

the velocity profile is thus like the bottom diagram in the above Figure 4b. Once the boundary layer has reached the center of the pipe the flow it is said to be fully developed. (Note that at this point the whole of the fluid is now affected by the boundary friction.) The length of pipe before fully developed flow is achieved is different for the two types of flow. The length is known as the entry length.

      -Laminar flow entry length is 120 diameters
      -Turbulent flow entry length is 60 diameters

5.0 Friction Factor

In fluid dynamics, the Darcy friction factor formulae are equations – based on experimental data and theory – for the Darcy friction factor, **f**. The Darcy friction factor is a dimensionless quantity used in the Darcy–Weisbach equation for the description of friction losses in pipe flow as well as open channel flow. It is also known as the Darcy–Weisbach friction factor or <u>Moody friction factor</u> as shown in Figure 5 below.

5.1 Friction Losses in Pipes

Friction losses are a complex function of the system geometry, the fluid properties and the flow rate in the system. By observation, the head loss is roughly proportional to the square of the flow rate in most engineering flows (fully developed, turbulent pipe flow). This observation leads to the Darcy-Weisbach equation in Eqn. (2) for head loss due to friction:

$$\Delta h_f = f \frac{L}{D} \frac{V^2}{2g} \quad (2)$$

<u>which defines the friction factor,</u> **f. f** is sensitive to moderate changes in the flow and varies as function of Reynolds Number for fully turbulent flow. Thus, it is often useful to estimate the relationship as the head being directly proportional to the square of the flow rate to simplify calculations.

$$f = f\left(\text{Re}, \frac{\varepsilon}{D}, \text{pipe cross-section}\right), \quad \text{Re} = \frac{VD}{\nu}$$

-Reynolds Number (Re) is the fundamental dimensionless group in viscous flow (Velocity x Length Scale (D for pipes) divided by Kinematic Viscosity).
-Relative Roughness (**ε**) relates the height of a typical roughness element to the scale of the flow, represented by the pipe diameter, D.

-Pipe Cross-section is important, as deviations from circular cross-section will cause secondary flows that increase the pressure drop. Non-circular pipes and ducts are generally treated by using the hydraulic diameter in Eqn. (3), in place of the diameter, and treating the pipe as if it were round.

$$D_H \equiv \frac{4A}{P} = \frac{4 \text{ x the cross - sectional area}}{\text{the wetted perimeter of the pipe}} \quad (3)$$

For <u>laminar flow</u>, the head loss is proportional to velocity rather than velocity squared, thus the friction factor is inversely proportional to velocity as shown in Eqns. (4) and (5):

$$\text{Circular Pipes:} \quad f = \frac{64}{Re} \quad (4)$$

$$\text{Non - Circular Pipes:} \quad f = \frac{k}{Re}, \quad (5)$$

Geometry Factor k

Square   56.91
2:1 Rectangle   62.19
5:1 Rectangle   76.28
Parallel Plates 96.00

The Reynolds number must be based on the hydraulic diameter. <u>For turbulent flow</u>, Colebrook found an implicit correlation in Eqn. (6) for the friction factor in round pipes:

$$\frac{1}{\sqrt{f}} = -0.869\ln\left(\frac{\varepsilon/D}{3.7} + \frac{2.523}{Re\sqrt{f}}\right) \quad (6)$$

The familiar <u>Moody Diagram</u> shown in Figure 5 below is a log-log plot of the Colebrook correlation on axes of friction factor and Reynolds number, combined with the f=64/Re result from laminar flow.
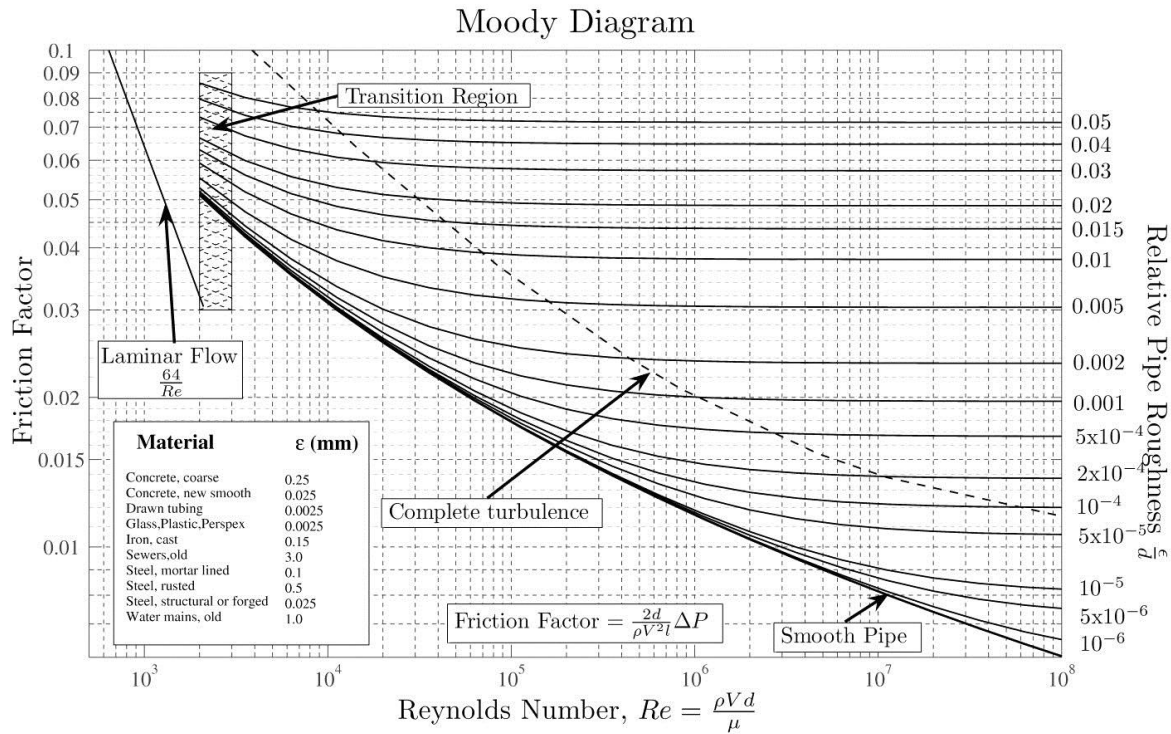
Figure 5 - Moody Diagram for Friction Factor for Pipes

6.0 Drag Reduction in Polymers

The application of drag reducing additives (DRAs) is used to reduce system energy requirements, reducing pipe diameter, or increasing flow rate, etc. The first famous application of DRAs was in transport of crude oil in the trans-Alaska (TAPS or Alyeska) Pipeline in 1979. The pipeline is 800 miles long and is 48-inch in diameter. After injecting a concentrated solution of a high molecular weight polymer downstream of pumping stations at homogeneous concentrations as low as 1 ppm, crude flow throughput was increased by up to 30%.

Drag reducing polymers can be used in district heating or cooling systems (DHC). These systems provide or remove heat in buildings or a district by a recirculation of water heated or chilled at a central station. The water recirculation energy requirements make up about 15% of the total energy for a DHC. In order to decrease this large proportion of the energy requirements, it is desirable to reduce friction in the water recirculation system. Thus, drag reduction by polymers is an alternative choice to meet this purpose. Polymer DRAs can reduce pumping energy

requirements by 50–70%. The effectiveness depends on the kinds of additives used and the layout of the primary system. The savings by DRAs are greater if the pipeline has fewer branches or the pipelines are longer or the number of fittings (e.g., valves, elbows, etc) is relatively small.

Besides DHC applications, a novel application of polymer DRAs is preventing flow-induced localized corrosion (FILC). Polymers can not only reduce wall shear stresses at a constant Reynolds number, but also suppress turbulent eddies near the wall, whose repeated impacts cause intermittent stresses on the wall, leading to mechanical damage to the surface material.

6.1. Drag Reduction Characteristics in Polymer Solutions

Polymer solution drag reduction has been studied extensively in both aqueous and hydrocarbon media by many researchers. Some important characteristics of polymer drag reduction are summarized below.

6.1.1. Drag Reduction Types

Early work proposed two types of drag reduction in dilute polymer solutions. Very dilute solutions display an onset drag reduction that is in the fully developed turbulence region. This drag reduction only occurs above an onset Reynolds number, where the friction factor decreases below that for ordinary Newtonian turbulent flow. At Reynolds numbers less than the onset value, no drag reduction occurs. Thus, there is an onset shear stress and an onset shear rate.

For more concentrated polymer solutions, drag reduction occurs by extension of the laminar region. In this case, onset conditions are reached at low Reynolds numbers, that is, the laminar-turbulent transition is not observed and laminar-like behavior is extended to an extended laminar region, in which the friction factor is lower than that of ordinary Newtonian turbulent flow at the same Reynolds number as shown in Figure 5. Drag reduction here and also in onset drag reduction increases with flow rate until a critical wall shear stress is reached , at which the rate of polymer degradation in the wall region exceeds the rate at which polymer is replenished in this region and drag reduction diminishes.

The main difference between these two kinds of drag reduction is the region where drag reduction occurs. The former begins in the fully developed turbulent region. The latter is observed in the extended laminar region.

6.2. The Onset Phenomenon

For flow with polymer solutions, drag reduction does not occur until a certain wall shear stress or shear rate is exceeded. This is the onset phenomenon. Its mechanisms are the stretching of the polymer molecules in the extensional flow regions or the entanglement of many molecules reaching the size of turbulent eddies. Researchers found the onset shear stress was not dependent on pipe diameter. The onset Reynolds number is proportional to the 8/7th power of the pipe diameter. So the same polymer solution has different onset Reynolds numbers in different pipe diameters, reflecting the onset shear stress or shear rate. Therefore, it is more appropriate to define the onset phenomenon using the onset shear stress or shear rate rather than the onset Reynolds numbers which are a function of polymer parameters. It decreases with increase of molecular weight of the polymer molecules.

In predicting the onset phenomenon, researchers proposed three different hypotheses, namely, length scale, time scale, and strain energy storage models. The length scale model proposed that onset occurs when the ratio of the polymer length scale (radius of gyration) to the turbulent length scale reaches a certain value. In the time scale model, the onset occurs when the time ratio (or the Deborah number (De) which is a dimensionless number, often used in rheology to characterize the fluidity of materials under specific flow conditions. It is based on the premise that given enough time even a solid-like material will flow) is around unity. De is defined as the ratio of polymer relaxation time and the turbulent time scale. Most researchers preferred the time scale model because polymer size is several orders of magnitude smaller than the turbulent eddy size at onset, while the polymer relaxation time is about the same order as the turbulent time scale. In the strain energy storage model, it is proposed that the polymer molecules stored energy when they are strained by high shear stress near the wall, and released it by relaxation when transported to the low shear region at the core of the flow. The onset of drag reduction occurs when strain energy convection is comparable to energy diffusion. This model as well as the time scale model predicts a decrease in onset Deborah number as polymer concentration increases which has been confirmed by experiment.

The addition of small amounts of long-chain polymers to flowing fluids produces large effects on a wide range of phenomena such as the stability of laminar motion, transition to turbulence, vortex formation and break-up, turbulent transport of

heat, mass and momentum, and surface pressure fluctuations. The feature that has attracted the most attention is the <u>reduction of friction drag by up to 80%</u> when only a few tens of parts per million by weight of long-chain polymers are added to a working fluid such as water.

6.3 The Mechanisms of Drag Reduction

The mechanisms of drag reduction are defined by the interaction of two fields, namely polymers in solution and turbulent flows. How this interaction occurs can be explained by considering the flow in a pipe.

Many aspects of the drag reduction in pipes occur because of several changes in the flow: changes in the mean velocity profile, alteration of turbulent stresses and the reduction of the correlation between the stream wise and radial velocity fluctuations, reduction of high-order moments of fluctuations, modification of the turbulent structure, and especially at the wall. Attention will be focused on the two first-order effects, both of which are explained best with reference to Figure 6.
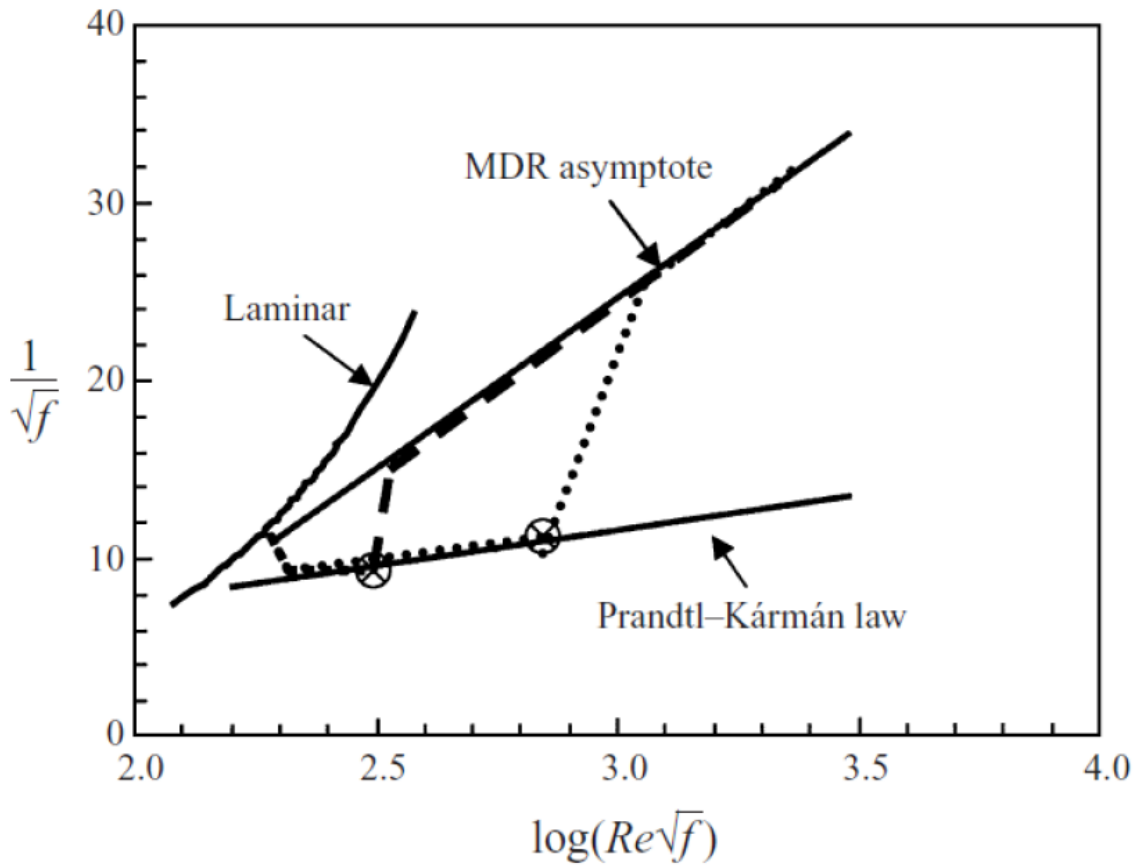
Figure 6 Drag Reduction Regions for Pipe Flows

Figure 6 shows the conventional friction drag for pipe flows in the so-called Prandtl-Karman (P-K) coordinates. The friction factor, *f,* in Eqn. (7) is related to the pressure drop p across a length L of the pipe of radius R as

$$f = \frac{\Delta p}{\rho V^2} \frac{R}{L}. \qquad (7)$$

The pressure drop in the turbulent pipe flow of a Newtonian fluid follows the straight line denoted as the Prandtl-Karman law. A dilute solution of a polymer follows the P-K law up to a certain Reynolds number but abruptly departs from it at higher Reynolds numbers. <u>This point of departure is the start of the onset of drag reduction</u>. The onset occurs at a lower Reynolds number if the polymer concentration is increased, as shown qualitatively in the figure.

As the Reynolds number is increased, the drag reduction curves of Figure 6 will

ultimately merge with the line denoted as the `MDR asymptote'. It should be noted that increasing the polymer concentration does not lead to drag reductions in excess of this asymptote (which is insensitive to polymer species, molecular weight, or the type of polymer).

6.4 Concentration Effect

Generally speaking, drag reduction at a fixed velocity increases as polymer concentration increases until a certain concentration is reached. This concentration is usually called the saturation concentration. Above this concentration, drag reduction decreases. The initial increase in drag reduction with concentration increase is probably due to the increasing number of polymer molecules which cause the damping of more turbulent eddies in the flow field. The decrease in drag reduction after the saturation concentration is caused by an increase in solution viscosity.

6.5 Molecular Weight Effect

When molecular weight is less than 105, polymers are generally ineffective for drag reduction. For a given concentration and Reynolds number, drag reduction increases with increasing average molecular weight. It has been concluded that for the same polymer species an increase in molecular weight resulted in a decrease in onset shear stress and a decrease in the amount of polymer required to obtain the same level of drag reduction. Researchers have studied the molecular weight effect, using a modern gel permeation chromatography technique, which allows separation of the high molecular weight portions of the polymer. Their results confirmed the conclusion that the highest molecular weight fraction of polymer primarily determined the onset shear stress. They confirmed this conclusion quantitatively by observing that, when 2.5 ppm of a high molecular weight polystyrene (7.1 million) and 25 ppm of a low molecular weight polystyrene (1.8 million) were mixed, the drag reduction was almost identical to that using the high molecular weight sample alone. However, when the amount of low molecular weight polymer was increased to 100 ppm, the addition of the low molecular weight polymer made a significant contribution to the drag reducing effectiveness of the mixture. It was also suggested that good drag reduction can be achieved by low molecular weight polymers if their concentrations are high enough. This is how Cold-Plus™ polymers achieve such large drag reduction.

6.6 Molecular Conformation and Solvent Effect

Researchers found that linear polymers without branches are more effective drag reducers than other polymers. Polytetrafluoroethylene (PTFE) (as used by Cold-Plus™) and polydimethylsiloxane (PDMS), the two most flexible polymers, are the most effective polymer drag reducers. The effectiveness of polymer solutions depends on the stretching of individual molecules by the stresses in the flow.

The molecular conformation can lead to different types of drag reduction. For dilute less flexible randomly coiled polymers, a transition from laminar to turbulent flow is usually observed and onset occurs in the turbulent region. On the other hand, flexible polymers usually do not show a clear transition. Drag reduction occurs in the extended laminar region. This is probably because the onset shear rate or shear stress has already been reached in the laminar or transition zone.

6.7. Diameter Effect

For Newtonian fluids in the turbulent region, each Reynolds number corresponds to a specific friction factor as shown in Figure 6. However, for viscoelastic fluids at a given Reynolds number, the same polymer solution in different diameter pipes gave different values of the friction factor. In general, the drag reduction observed in large pipes is smaller than that obtained in small pipe systems because of lower wall shear stresses and shear rates.

Viscoelastic theories predict a 1.1-power dependency of critical solvent Reynolds number on diameter in Newtonian solutions .It was noted that the critical solvent Reynolds number varies with about the first power of diameter, in good agreement with the predictions. Thus, for a solution which has a critical solvent Reynolds number for drag reduction of 100,000 in a 2-centimeter diameter tube, the critical solvent Reynolds number for an 8-centimeter tube would be about 400,000. Thus the larger the diameter, the higher the critical Reynolds number needed to initiate drag reduction.

7.0 Heat Transfer Characteristics

The heat transfer that takes place in the pipe occurs at the fluid-wall interface. At that point, the heat transfer is directly related to the flow in the pipe. When the flow becomes turbulent, the turbulent mixing of the fluid causes the fluid next to the wall to mix with the free-stream fluid and provide heating to the fluid and increasing the fluid temperature, assuming the pipe temperature is greater than the

free stream fluid temperature. The overall effect of this interaction is to increase the heat transfer from the wall to the fluid. However, if the fluid is at a higher temperature than the wall of the pipe, than the heat transfer will take place in the opposite direction.

The heat transfer correlation that is used to represent the heat transfer for flow in a pipe is the Nusselt Number. The Nusselt Number for pipe flow is given in Equation 8 as:

$$Nu = hD/k = 0.023 \ Re^{0.8} \ Pr^{0.4} \qquad (8)$$

where h = heat transfer coefficient
      D = pipe diameter
      k = thermal conductivity
      Re = Reynolds Number

As the flow goes from laminar to turbulent, the heat transfer coefficient, h, will increase as the Reynolds Number increases as shown in Figure 7.
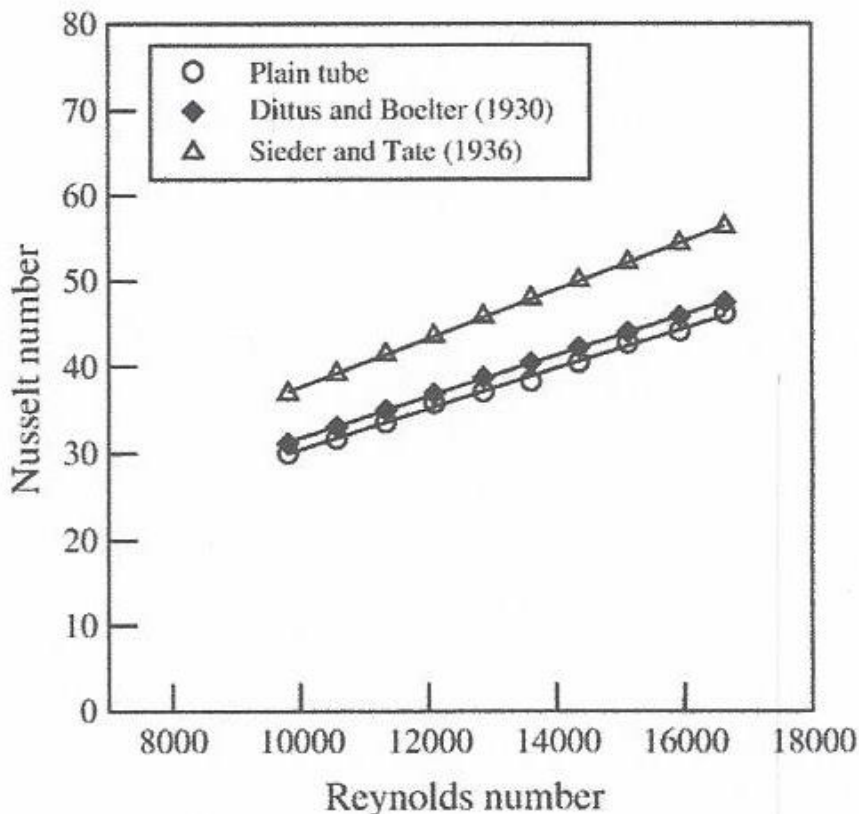

Figure 7 Nusselt Number vs. Reynolds Number for flow in a pipe

After determining the heat transfer coefficient, h, from Equation 8, the heat transfer from the wall to the fluid (as shown in Figure 9) can be obtained from the following Equation 9,

$$Q = h \, A \, (T_w - T_f) = Nu \, (k/D) \qquad (9)$$

where
T_f = fluid temperature
T_w = wall temperature

Since the heat transfer to the wall Q is directly proportional to the heat transfer coefficient, h (as shown in Figure 8), and Nusselt Number, then the heat transfer increases with Reynolds Number as shown in Figure 7. A 14% change in Reynolds Number will result in approximately a 25% increase in Nusselt Number and heat transfer.
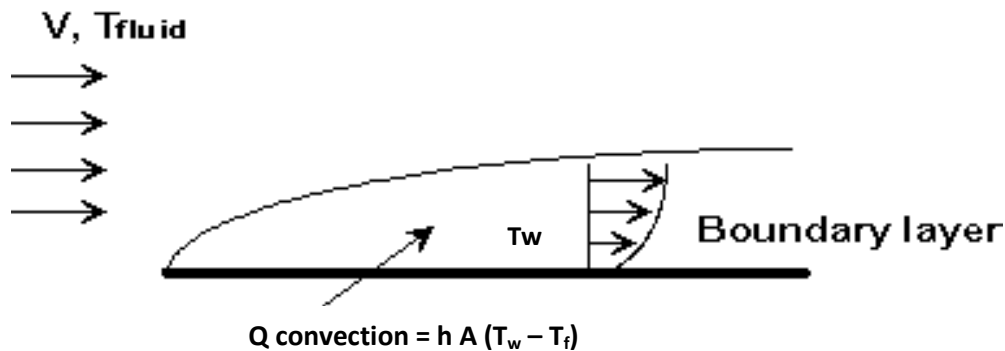

Figure 8 - Thermal Boundary Layer

The heat transfer will be affected by the pipe surface roughness and how well the polymers adhere to the inside of the pipe surface.

7.1 Boiling Heat Transfer

A boiling process dissipates thermal energy within the evaporator when the refrigerant is heated (boils) as it traverses through the evaporator coil and withdraws heat from the cooled space. The heat energy in the surrounding cooled space is dissipated into the liquid by the transfer of heat from the cooled space through the walls of the evaporator coil to the refrigerant within the evaporator coil.

As energy (heat) flow increases from a solid surface, such as the inside wall surface of the evaporator coils, as shown in Figure 8, to a liquid, such as the refrigerant liquid flowing in the evaporator coils, the fluid temperature increases and a point is reached (referred to as the nucleate boiling point) where vapor bubbles form (i.e., the fluid begins to boil) on the inside wall surface of the coils. These bubbles form at preferred sites. If the liquid temperature is below a saturation temperature, the vapor bubbles collapse soon after formation. However, as the liquid temperature increases as a result of the transfer of more heat energy into the liquid (from the surrounding space to be cooled) the bubbles become more numerous and are released from the formation sites and carried into the fluid as shown in the Figure 9 below. This process is referred to as nucleate boiling
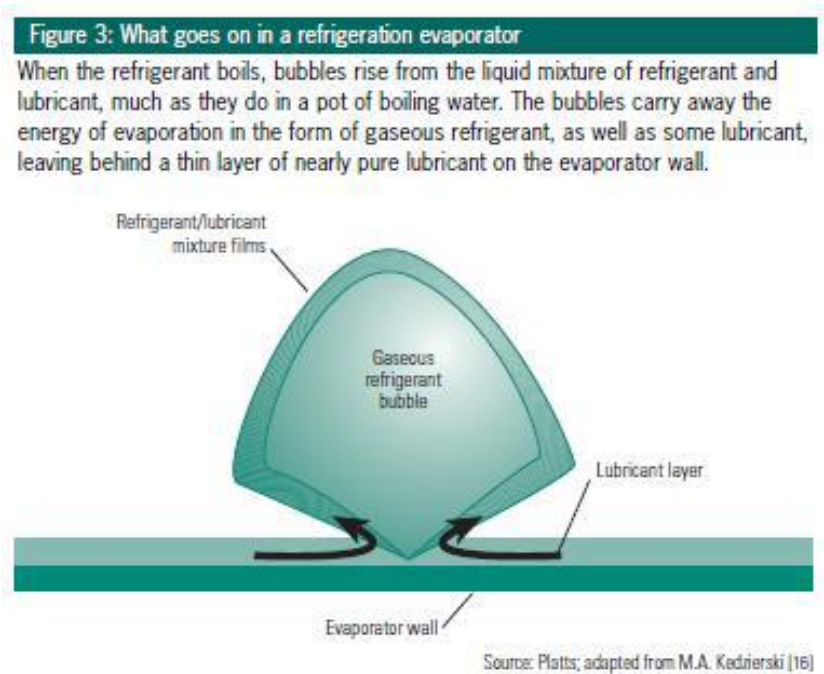

Figure 3: What goes on in a refrigeration evaporator
When the refrigerant boils, bubbles rise from the liquid mixture of refrigerant and lubricant, much as they do in a pot of boiling water. The bubbles carry away the energy of evaporation in the form of gaseous refrigerant, as well as some lubricant, leaving behind a thin layer of nearly pure lubricant on the evaporator wall.

Refrigerant/lubricant mixture films

Gaseous refrigerant bubble

Lubricant layer

Evaporator wall

Source: Platts; adapted from M.A. Kedzierski [16]

Figure 9 - Bubble Formation

Figure10 illustrates a typical boiling curve, including a heat flux plot (total heat flow divided by area (q/A) and a heat transfer coefficient (h) curve. The heat flux and heat transfer coefficient are plotted on the ordinate against a temperature difference between the heated surface and the saturation temperature of the liquid on the abscissa. Five curve regions are identified: natural convection (free-surface evaporation), nucleate boiling, transition boiling, and film boiling. The first two regions are commonly referred to as pool boiling regions.
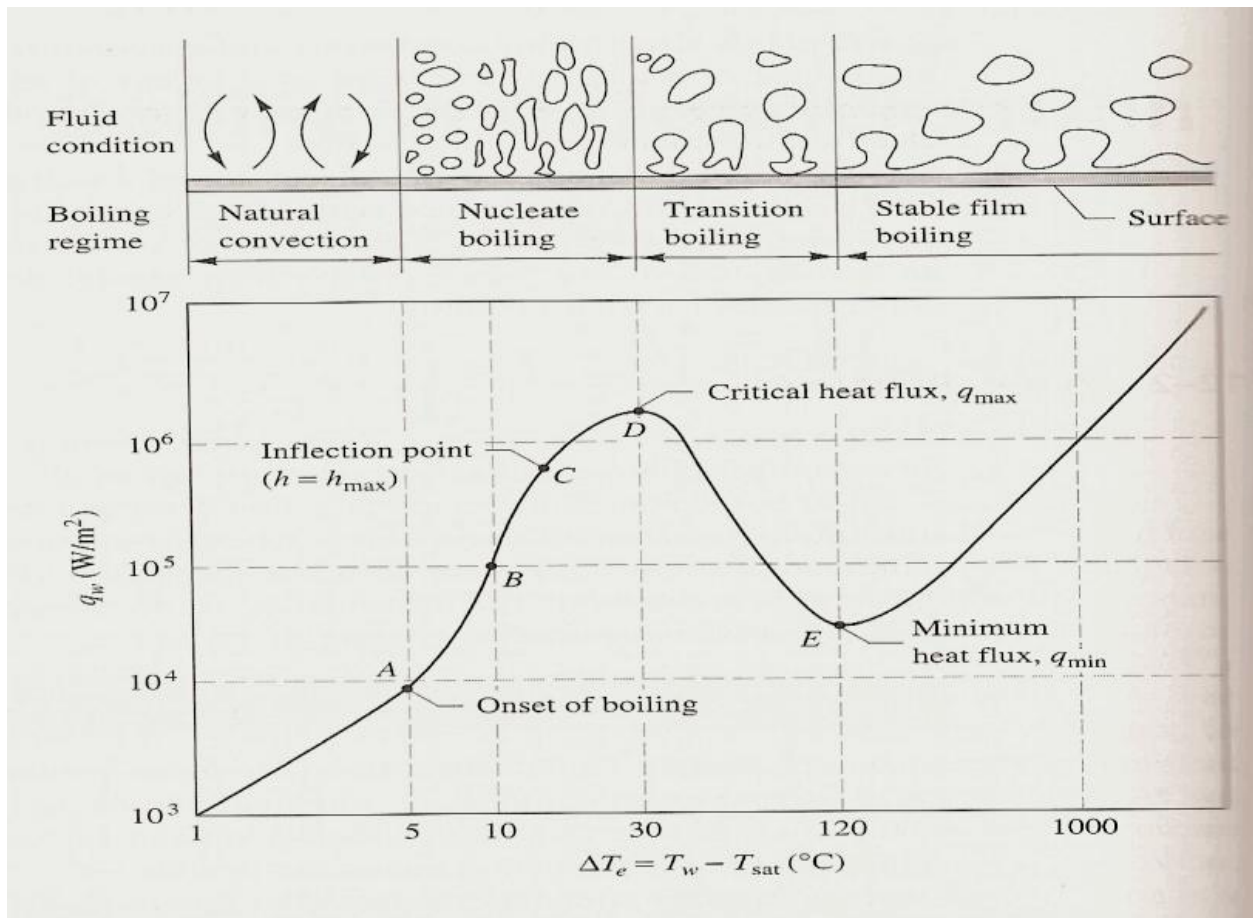
Figure 10 Boiling Curve

For low values of heat flux the heat flux plot is a straight line, only a portion of which is shown in Figure 10. In this regime, as applied to a refrigeration cycle, heat transfer to the refrigerant fluid is primarily via conduction from the ambient air of the cooled space through the interface with the fluid (e.g., through the walls of the evaporator coil, as shown in Figure 8). The heat is then transferred by forced convection throughout the fluid. The fluid evaporates at an open or free surface when the fluid temperature reaches its boiling point. The heat flux in this regime is relatively low compared to the other regions of the curve and as expected the associated heat transfer coefficient (h) as depicted by the curve is also relatively low. As the refrigerant in the evaporator tubes absorbs heat, it moves toward the superheat region. In this regime a convective heat transfer coefficient can be

determined, as outlined in Section 7.0, and used to calculate the heat transfer rate. The increase in heat flux as the refrigerant moves through the evaporator tubing is determined by the given flow Reynolds Number as given by Equations (8) and (9).

In a region of the curve between curve points A and B the temperature of the evaporator coil/ambient air interface continues to increase by conduction from the space to be cooled, causing individual bubbles to form on the heated surface, i.e., the inside wall surface of the evaporator coil. The appearance of the bubbles increases the heat flux from the heated surface to the fluid, as indicated by a steady increase in the heat flux curve in the Figure 10. The heat transfer coefficient also increases in this region of the curve.

From curve points B to C (a relatively linear segment of the plot) bubble columns form in the column as a result of the increased heat transferred to the fluid refrigerant as indicated by the increase in the heat flux and the heat transfer coefficient. At the point C the heated surface is crowded with vapor bubbles and the rate of increasing heat transfer begins to decline. The point C denotes a departure from nucleate boiling (DNB).

If the heat flux is raised further to a maximum heat flux point, q_{max}, an insulating film comprising a blanket of vapor bubbles begins to form on the heated surface. As the excess temperature increases along the abscissa the insulating film lowers the heat flux. This film boiling is unstable in the region between points D and E, a stable film region beyond point E and a region where radiation from the surface affects the insulating film. As illustrated, the temperature difference changes rapidly in this region to a curve endpoint. For any practical cooling system it is desired to maintain the operating point at or below the point C thereby avoiding a departure from nucleate boiling region.

7.2 Nucleate Boiling

A process referred to as superheated boiling occurs in the nucleate boiling region as the temperature of the fluid within cavities on the surface of the heat transfer interface exceeds the boiling point. The liquid in the immediate vicinity of these cavities is super-heated. These surface conditions affect the shape of the boiling curve for a given interface material and liquid.

Superheated boiling involves a phase transition (liquid to gas) in a small but stable region of the liquid. The liquid (fluid) boils as the gas bubbles are formed (see Figure 9) in the micro cavities of the surface on the surface of the refrigerant

tubing. The bubbles grow until they reach a critical size; then they separate from the surface and enter the liquid stream, transferring heat energy from the cooled space to the refrigerant fluid. This process denotes the beginning of the boiling process as depicted at point A on the boiling curve of Figure 10.

Three parameters affect the superheated boiling heat transfer performance of a wall or plate and a surface that is exposed to a fluid (such as the inside surface of an evaporator coil that is exposed to the refrigerant). The first parameter is the liquid superheat required to initiate boiling. To initiate the nucleation process, the wall, the surface and then the liquid (e.g., the refrigerant) must reach an incipient superheat condition. The incipient superheat condition is inversely proportional to a volume of any vapor or gas trapped within voids of the wall or irregularities in the surface, because the trapped gas lowers the thermal conductivity of the wall/surface.

The second performance parameter is the surface-to-liquid heat transfer coefficient (see the curve of Figure 10) when the liquid is in the nucleate boiling regime since this is the desired condition for boiling. The heat transfer coefficient is affected by the properties of the surface (including irregularities and particles on the surface) and is directly proportional to the density of the nucleation sites on the surface. Increasing the heat transfer coefficient leads to higher heat flux as more heat is transferred from the surface to the boiling fluid. In the evaporator, increasing the heat transfer coefficient increases the heat flux and therefore causes more heat to be removed from the cooled space to the circulating refrigerant.

The third boiling performance parameter is the critical heat flux (CHF), defined as the highest heat flux that can be removed without exposing the surface to film boiling. Generally this value is represented by the point q_{max} in Figure 10.
An air conditioning fluid (refrigerant and lubricating oil) or a treatment added to the fluid of an existing air conditioning system, comprised of particles that modify the surface characteristics of the system's heat exchangers, coat the inside surface of the evaporator tubes and the condenser tubes. The particles are deposited on the inside surface of these tubes as the fluid circulates through the system.

The enhanced heat transfer, and thus increased system efficiency, is due to one or more of the following effects caused by the deposited particle layer: increasing the number and/or density of superheated boiling sites, increasing the heat transfer coefficient of the coil inside surface (increasing heat transfer to the fluid), increasing the critical heat flux (i.e., increasing the heat flux, q_w, for the same excess temperature, ΔT, and lowering the temperature at which the liquid incipient

23

superheat condition occurs (i.e., the particulate layer produces the same heat flux at a lower temperature or a higher heat flux for the same temperature). Alone or in combination these conditions increase the heat transfer efficiency from the space to be cooled to the system fluid.

The particles in the fluid or fluid treatment adhere to or deposit on inside surfaces of the evaporator coil, forming a particle layer that comprises superheated boiling-site cavities. The particles can be referred to as cavity-generating particles. As used here, "cavity-generating particles" means any particles capable of forming depressions in a particle layer having a width from about 0.5 μm to about 10 μm. Such depressions are suitable for promoting superheated boiling. Preferred particles include crystals, flakes and randomly shaped particles, but could also include spheres or any other shaped particle that provide suitable cavities. The particles can comprise up to about 25% by weight of the fluid circulating in the AC system (the fluid typically comprising the refrigerant and the lubricant).

The superheated boiling sites may present pointed features, such as the cavity edges or areas of roughness in the particle layer,that promote the formation of bubbles. The formation of bubbles from more boiling sites increases heat transfer from the heat transfer surface into the fluid. Within the fluid the heat is transferred by forced and natural convection.

7.3 Polymer and Particle Effects

The cavity generating particles comprise PTFE and Cold-Plus™ particles that tend to impregnate or embed in surfaces they contact. These particles impregnate the inside surface of the evaporator coils (and condenser coils) to form a new heat transfer surface with the fluid refrigerant circulating within the coils. In addition to forming the advantageous superheated boiling sites that improve the boiling performance of the refrigeration system, the particle layer also affects the conduction and convection heat transfer characteristics between the space to be cooled and the refrigerant circulating within the evaporator coil. The irregular surface characteristics (e.g., the nucleating site cavities) formed as the particles deposit on the surface increase the surface area along the fluid/surface interface, increasing the fluid-surface contact and thereby increasing the conductive heat transfer from the cooled space to the fluid. With increased heat transfer to the surface, the superheated boiling sites reach a higher temperature. This temperature is a defining driver for boiling (forming bubbles) the fluid at the nucleating sites.

Further, since the particles have a higher heat transfer coefficient than the fluid, they increase the thermal conductivity (and thus the heat flux) from the space to be cooled into the refrigerant fluid. This effect also raises the temperature of the superheated boiling sites, improving the boiling characteristics of the fluid. Since any voids formed in the particle layer during deposition tend to fill with fluid and thereby decrease the thermal conductivity of the particle layer (since the fluid has a lower thermal conductivity than the particle material), particles that are shaped and sized to fit the voids tend to reduce the formation of fluid-filled voids and are therefore desired. Also, as illustrated by the film boiling region of the boiling curve in Figure 10, to the extent the fluid-filled voids form a fluid film, the heat transfer characteristics are drastically reduced. Generally, relatively smaller particles in the particle layer increase the heat transfer rate more than larger particles, as the smaller particles tend to pack more closely, reducing the number of fluid-filled voids.

The increased heat transfer rate observed appears to be at least partially caused by the relatively high thermal conductivity of the particles and their single-phase heat transfer coefficient near the heat transfer sites of the interior walls of the evaporator coil. Thus the heat is transferred from the space to be cooled to the refrigerant by conduction through the material of the evaporator coil walls and the particulate layer formed as described above and by convection as the heat enters the air conditioning fluid. As heat is transferred into the fluid the fluid temperature increases and boiling occurs, i.e. bubbles form at the boiling sites. It is therefore desired to utilize particles that can both form the superheated boiling sites and present a relatively high thermal conductivity. Further, the particles increase the heat transfer process in the refrigerant by improving the mixing and turbulence of the fluid as it traverses through the system.

Using PTFE with the Cold-Plus™ particles (in the micron or nano size range), the particles are combined with other fluids (e.g., a suspension agent) to create a liquid formulation that is added to the circulating fluid of an operating AC system. Due to its lubricating properties (due primarily to the PTFE), thermal conduction properties and propensity to create boiling sites on the inside surface of the evaporator coil, the particles increase the efficiency, including heat transfer efficiency, of the air conditioning system, thereby reducing its power consumption (the energy required to operate the compressor). Tests have indicated a 14% to 17% system energy reduction. The beneficial operating features are realized within minutes of adding the particles to the fluid. The particles may also extend the life of system components and reduce the requirements for regular maintenance.

The PTFE also tends to adhere to the compressor surfaces, lowering the friction between the surfaces as outlined in Section 3 above. The lower friction allows the compressor to operate more efficiently and reduces the power consumption of the air conditioning system.

The density of the particles and the viscosity of the material in which the particles are suspended within the circulating fluid also affect the heat transfer properties of the system. A higher particle density may permit deposition of a sufficient number of particles on the heat transfer surfaces to reduce formation of voids in the particle layer.

The Cold-Plus™ particles are added to a lubricant that is added to the refrigerant. A lubricant or other additive (that is, an additive carrying the particles in suspension) having a viscosity greater than the viscosity of the refrigerant promotes formation of a particulate boundary layer on the heat transfer surface should be used, allowing the particles to offer the beneficial heat transfer characteristics described above in Section 7.0.The additive may replace less viscous lubricant on the heat transfer surfaces, causing formation of the particle layer on those surfaces. If the refrigerant and the additive are substantially chemically similar, the layer of particles may not form as desired. The miscibility of the additive also affects the deposition of particles on the heat transfer surfaces and thus the heat transfer characteristics of those surfaces.

The surface tension of the particles also affects the heat transfer characteristics. As the surface tension of the particles increases the heat transfer rate is reduced, reducing the boiling initiation point.

7.4 Cold-Plus™ Applications

The Cold-Plus™ treatment can be added to the fluid (refrigerant and lubricant) of an operating HVAC (heating ventilating air conditioning) system and to the fluid of an operating packaged AC system. Alternatively, the treatment material is included with the fluid at the point of manufacture and injected into the air conditioning system when it is charged with refrigerant.

Although the beneficial effects associated with increased heat transfer due to the characteristics of the particle surface layer have been described in conjunction with an air conditioning system evaporator, the process is also applicable to the AC system condenser. The heat transfer improvements described above for the evaporator side of the refrigerant cycle also present for the condenser side. It is

known however, that the properties of the fluid differ as the fluid state moves from the superheat to the saturated mixture to the compressed liquid region (i.e., from point 2 to point 5 of Figure 10).

This process can also be applied to cooling elements or components that are to be maintained below a predetermined temperature for proper operation. Packaged electronic components are cooled by immersion in a cooling fluid or by passing cooling fluid proximate the external surfaces thereof. By treating the surfaces according the present process the heat transfer characteristics are improved and the cooling efficiency increased.

Although described in the context of an air conditioning (cooling) system, these processes are applicable to any refrigeration or cooling system. The lubricant can be used with other types of cooling systems, including automotive, residential and building air conditioning systems, further including refrigerators, heat pumps, air conditioning systems and chillers. The material is also suitable for use with systems for cooling semiconductor devices and electronic components.

8.0 Summary

The information given in this report is to provide insight into why the addition of the Cold-Plus™ PTFE polymer reduces drag and increases heat transfer in refrigeration systems. The friction drag is directly related to the flow Reynolds Number , and as the Reynolds Number increase with the addition of the PTFE additive, the resulting flow friction, f, is reduced. This reduction in flow friction results because of the way the polymer interacts with flow boundary layer and turbulence in the pipes. As a result of this reduction in friction and drag, the system power requirements are reduced and there is an overall improvement in the system efficiency.

Also, because of the turbulent mixing of the fluid at the pipe wall, the fluid temperature will be increased. As the Reynolds Number of the flowing fluid is increased, the Nusselt Number will increase and result in an increase in the heat transfer from the wall to the fluid. This increase in heat transfer will increase the performance of the A/C system.

The PTFE and the Cold-Plus™ particles are combined with other fluids to create a liquid formulation that is added to the circulating fluid of an operating AC system. Due to its lubricating properties, thermal conduction properties and propensity to create boiling sites on the inside surface of the evaporator coil, the particles

increase the efficiency, including heat transfer efficiency, of the air conditioning system, thereby reducing its power consumption (the energy required to operate the compressor). Tests have indicated a 14% to 17% system energy reduction. The beneficial operating features are realized within minutes of adding the particles to the fluid. The particles may also extend the life of system components and reduce the requirements for regular maintenance.

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