

A
Thesis
On

**HEAT TRANSFER TO NEWTONIAN AND NON-NEWTONIAN FLUID
IN AGITATED VESSEL**

**SUBMITTED IN PARTIAL FULFILLMENT FOR THE REQUIREMENT FOR THE
AWARD OF THE DEGREE OF**

**DOCTOR OF PHILOSOPHY
IN
MECHANICAL ENGINEERING
BY**

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2017

DECLARATION

I, **Ansar Ali Sk**, declare that the work presented in this thesis entitled **Heat Transfer to Newtonian and Non-Newtonian Fluid in Agitated Vessel**’ submitted to the **Department of Mechanical Engineering**, in the Faculty of **Engineering and Technology**, **Sam Higginbottom Institute of Agriculture, Technology and Sciences, Deemed-to-be University, Allahabad**, for the award of the **Doctor of Philosophy** degree in **Mechanical Engineering** is an original work. The optimum results have been obtained with standard data. The report embodies result of original work and studies carried out by myself.

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TABLE OF CONTENTS

TITLE	PAGENO.	
TITLE PAGE		
UNDERTAKING	i	
CERTIFICATE OF ORIGINAL WORK	ii	
STUDENT ADVISORY COMMITTEE CERTIFICATE	iii	
EVALUATION COMMITTEE REPORT	iv	
ABSTRACT	v- vi	
ACKNOWLEDGEMENTS	vii	
LIST OF TABLES	viii- ix	
LIST OF FIGURES	x- xii	
NOMENCLATURE	xiii-xv	
CHAPTER	TITLE	PAGE NO.
CHAPTER –1	INTRODUCTION	1-6
1.1	General Introduction	1
1.2	Rheological Classification of the Fluids	1-3
1.3	Heat Transfer in Stirred vessels	3-5
1.4	Objective	5-6
CHAPTER- 2	REVIEW OF LITERATURE	7-24
CHAPTER- 3	MATERIALS AND METHODOLOGY	25-61

3.1	Heat transfer from vessel wall to the agitated pseudoplastic fluids	25-26
3.2	Present Approach	26-30
3.3	Heat transfer from agitated Non-Newtonian fluid to helical coils	30
3.4	Laminar forced convection to Non-Newtonian fluid in coiled tube	31
3.5	Measurement of Rheological Properties	32
3.5.1	Viscometre	32-33
3.5.2	Procedure	34
3.5.3	Calibration	34
3.5.4	Determination of the pseudoshear curve	35-51
3.7	Experimental setup	52-56
3.8	Procedure	56-57
3.9	Determination of thermal conductivity	58
3.9.1	Apparatus	58-59
3.9.2	Procedure	60-61
3.10	Determination of specific heat	61
CHAPTER-4	RESULTS AND DISCUSSION	62-99
4.1	Results	62
4.1.1	Range of Variables	62
4.1.2	Heat Balance and Overall Heat Transfer Coefficient	63
4.1.3	Analytical Procedure for Individual Heat Transfer Coefficients, h_j , h_{oc} and h_{ic}	63-70
4.1.4	Non Isothermal Correction	70
4.2	Correlation	70
4.2.1	Heat Transfer from Jacket to Agitated Non-Newtonian Fluids	70-71
4.2.2	Effect of Prandtl Number	71

4.2.3 Effect of Reynolds Number	76	
4.2.4 Effect of Impeller diameter	76	
4.2.5 Heat Transfer from Agitated Non-Newtonian Fluids to the Helical Coil	77	
4.3 Comparison and Discussion	87-90	
4.3.1 Laminar Forced Convective Heat Transfer to Non-Newtonian Fluids in Helical Coil	90-91	
4.3.2 Effect of Dean Number N_{D2}	96-97	
4.3.2 Effect of Prandtl Number	97	
CHAPTER –5	SUMMARY AND CONCLUSIONS	100-143
REFERENCES		144-149
APPENDICES		150-161
LIST OF PUBLICATIONS		

ABSTRACT

The design engineer is very often faced with the problem of designing process vessels for handling fluids. Many of these equipments are needed for such varied operations as heating, cooling, mixing etc of non-Newtonian fluids. Heat addition into and heat removal from such fluid contained in an agitated vessel is common industrial application. These equipment, technically known as agitated vessel heat exchangers are either jacketed and / or equipped with helical coils and fluids are either agitated in the vessel or circulated through the coils immersed in a fluid contained in the vessel.

Much is known about the heat and momentum transfer to Newtonian fluids being processed in these equipments whereas such information concerning non-Newtonian fluids is meager and needs further attention. In the present work, therefore, an attempt has been made to study the momentum and heat transfer characteristics of non-Newtonian fluids in agitated vessels and in helical coils and to clarify some of the anomalies existing in the previous investigations. The non-linear nature of shear stress-shear rate relationship of non-Newtonian fluids makes the solution of momentum and energy equations more complex. However, the knowledge of Newtonian behavior and correlation form a basis for non-Newtonian fluids, and therefore, Newtonian fluids have also been investigated. The present thesis embodied the resulting subject matter.

The importance of the problem, i.e. heat transfer to Newtonian and non-Newtonian fluids flowing through helical coils and agitated vessel, taken up for the present investigation and a brief introduction to non-Newtonian fluids are presented in chapter I.

The available literature on heat transfer in agitated vessels and coils is reviewed in chapter II. Large numbers of correlation for heat transfer to Newtonian and non-Newtonian fluids from vessel wall or coils presented by earlier workers differ from each other due to the use of different viscosity expressions in Reynolds and Prandtl groups.

Chapter III deals with the aim of this part of the investigation. The use of a viscosity expression corresponding to viscosity evaluated at the impeller tip, following the concept of a cylinder of diameter equal to that of impeller rotating in an infinite fluid, has been suggested. This is due to the fact that under turbulent flow conditions and for large gap between vessel wall and impeller, the vessel offers negligible effect on the fluid viscosity at the impeller tip.

Chapter IV deals with the result of experimental program for the measurement of rheological properties of non-Newtonian fluids.

Chapter V gives the summary of the experimental set up used for heat transfer measurements in coil as well as in the agitated vessel. A Newtonian fluid and four non-Newtonian fluids have been investigated. Experimental results of heat transfer to agitated fluids from vessel wall and to non-Newtonian fluids from vessel wall to non-Newtonian fluids flowing through helical coils in laminar fashion are discussed. The correlations obtained are compared with the results of some of the earlier investigations.

Lastly it contains conclusion with experimental data in tabular form.

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ANSAR ALI SK

LIST OF TABLES

Table No.	Name of the Table	Page No.
3.6.1-3.6.4	RHEOLOGICAL PRPPERTIES	42-51
5.1.A	List of fluids and range of variables studied for heat transfer from the vessel wall to the agitated fluids and agitated fluids to the coil.	102
5.1.B	List of fluids and range of variables studied for heat transfer to fluids flowing through heat transfer test coil.	103
5.2	Experimental data for heat transfer from jacket to vessel fluid and vessel fluid to coil (fluid: water)	104
5.3	Experimental data for heat transfer from jacket to vessel fluid and vessel fluid to coil (fluid: CMC-A)	105
5.4	Experimental data for heat transfer from jacket to vessel fluid and vessel fluid to coil (fluid: 2% CMC-A)	106
5.5	Experimental data for heat transfer from jacket to vessel fluid and vessel fluid to coil (vessel fluid : 4% CMC-A; coil fluid: 0.5% CMC-A)	107
5.6	Calculated variables for heat transfer from jacket to the agitated fluid (fluid: water)	108
5.7	Calculated variables for heat transfer from jacket to the agitated fluid (fluid: 1% CMC-A)	109
5.8	Calculated variables for heat transfer from jacket to the agitated fluid (fluid: 2% CMC-A)	110
5.9	Calculated variables for heat transfer from jacket to the agitated fluid (fluid: 4% CMC-A)	111
5.10	Comparison between experimental and calculated Nusselt numbers for heat transfer from vessel wall to the agitated fluids.	112-113
5.11	Calculated variables for heat transfer from agitated fluid to the coil (fluid: water)	114
5.12	Calculated variables for heat transfer from agitated fluid to the coil	115

	(fluid: 1% CMC-A)	
5.13	Calculated variables for heat transfer from agitated fluid to the coil (fluid: 2% CMC-A)	116
5.14	Calculated variables for heat transfer from agitated fluid to the coil (fluid: 4% CMC-A)	117
5.15	Comparison between experimental and calculated Nusselt numbers for heat transfer from agitated fluids to the coil.	118-119
5.16	Experimental data for heat transfer to fluid flowing through the coil (fluid: water)	120-122
5.17	Experimental data for heat transfer to fluid flowing through the coil (fluid: 0.5% CMC-A; vessel fluid : 4% CMC-A)	123
5.18	Experimental data for heat transfer to fluid flowing through the coil (fluid: 1% CMC-A)	124-125
5.19	Experimental data for heat transfer to fluid flowing through the coil (fluid: 2%CMC-A)	126-127
5.20	Calculated variables for pressure drop studies through heat transfer test coil ($D_t= 1.898$, $D_c = 34$; Fluids: water, 0.5% CMC-A, 1% CMC-A, 2% CMC-A)	128-133
5.21	Calculated variables for laminar flow heat transfer to fluids flowing through coil (fluid: water)	134-136
5.22	Calculated variables for laminar flow heat transfer to fluids flowing through coil (fluid: 0.5% CMC-A)	137
5.23	Calculated variables for laminar flow heat transfer to fluids flowing through coil (fluid: 1% CMC-A)	138-139
5.24	Calculated variables for laminar flow heat transfer to fluids flowing through coil (fluid: 2% CMC-A)	140-141
5.25	Comparison between experimental and calculated Nusselt numbers for laminar heat transfer to fluids flowing through coil.	142-143

LIST OF FIGURES

Figure No.	Name of the Figure	Page No.
1.2	Shear stress- shear rate diagram	02
3.5.1	Schematic diagram of capillary tube viscometer	33
3.5.4.A	Capillary flow diagram of 0.5% CMC-A	37
3.5.4.B	Capillary flow diagram of 1% CMC-A	38
3.5.4.C	Capillary flow diagram of 2% CMC-A	39
3.5.4.D	Capillary flow diagram of 4% CMC-A	40
3.5.4.E	Variation of consistency index with temperature	41
3.7.A	Schematic diagram of the experimental set-up	53
3.7.B-3.7.C	Photographs of Experimental Set up	54-55
3.9.1	Apparatus for thermal conductivity measurement	59
4.1.A	Modified Wilson plot for heat transfer from jacket to agitated 1% CMC-A SOLUTION, $1/U_j$ versus $1/N(3-n)/3$	65
4.1.B	Jacket side modified Wilson plot for water, $1/U_j$ versus $1/N^{2/3}$	66
4.1.C	Modified Wilson plot for heat transfer from jacket to agitated 2% CMC-A SOLUTION, $1/U_j$ versus $1/N(3-n)/3$	67
4.1.D	Modified Wilson plot for heat transfer from jacket to agitated 4% CMC-A SOLUTION, $1/U_j$ versus $1/N(3-n)/3$	68
4.2.A	Jacket to agitated fluid heat transfer correlation (effect of Prandtl number), $N_{Nuj} / N_{Rea}^{2/3}$ versus N_{Pra}''	71
4.2.B	Heat transfer correlation for agitated non-Newtonian fluids (effect of Reynolds number), $N_{Nuj} / N_{Pra}''^{2/3}$ versus N_{rea}''	72
4.2.C	Heat transfer correlation for jacket to agitated fluids: effect of agitator diameter, $N_{Nuj} / N_{Rea}''^{2/3} N_{Pra}''$ versus D_a / D_T	73

4.2.D	Heat transfer from jacket to agitated Newtonian and non-Newtonian fluids (effect of Reynolds number) $N_{Nu_j} / N_{Pra}^{1/3}$ $\left(D_a/D_T\right)^{0.1}$ versus N''_{Rea}	74
4.2.E	Comparison between experimental and calculated values of Nusselt number (heat transfer from jacket to agitated fluid)	75
4.2.F	Modified Wilson plot for heat transfer from coil to agitated water, $1/U_{oc}$ versus $1/N^{2/3}$	78
4.2.G	Modified Wilson plot for heat transfer to agitated 4% CMC-A solution, $1/U_j$ versus $1/N(3-n)/3$	79
4.2.H	Modified Wilson plot for heat transfer from coil to agitated 1% CMC-A solution, $1/U_j$ versus $1/N(3-n)/3$	80
4.2.I	Modified Wilson plot for heat transfer from coil to agitated 2% CMC-A solution, $1/U_j$ versus $1/N(3-n)/3$	81
4.2.J	Agitated fluid to coil heat transfer correlation (effect of Prandtl number), $N_{Nu_{oc}} / N''_{Rea}^{2/3}$ versus N''_{Pra}	82
4.2.K	Heat transfer correlation for coil to agitated fluid (effect of Reynolds number), $N_{Nu_{oc}} / N''_{Pea}^{2/3}$ versus N''_{Rea}	83
4.2.L	Heat transfer correlation for coil to agitated fluids (effect of agitator diameter) $N_{Nu_{oc}} / N''_{Rea}^{2/3} N''_{Pra}^{1/3}$ versus D_a/D_c	84
4.2.M	Heat transfer correlation for coil to agitated Newtonian and non-Newtonian fluids, $N_{Nu_{oc}} / N''_{Pra}^{1/3} \left(D_a/D_T\right)^{0.1}$ versus N''_{Rea}	85
4.2.N	Comparison between experimental and calculated Nusselt number (heat transfer from coil to agitated fluids)	86
4.3.A	Flow diagrams for 0.5, 1 and 2% CMC-A through heat transfer test coil, τ_w versus $8U/D_t$	92

4.3.B	Variation of friction factor with Reynolds number for water, 2, 1 and 0.5% CMC-A through heat transfer test coil, f_c versus N_{Re_2}	93
4.3.C	Laminar flow friction factor in helical coils as a function of Dean number for non-Newtonian fluids, f_c/f_{SL_2} versus N_{D_2}	94
4.3.D	Variation of N_{Nuic}/N_{Nuis} with N_{D_2} for heat transfer to non-Newtonian fluids	95
4.3.E	Variation of $\left(N_{Nuic}/N_{Nuis}\right) - 1$ with N_{D_2} for heat transfer to non-Newtonian fluids	96
4.3.F	Laminar flow heat transfer correlation for non-Newtonian fluids through coils: effect of Prandtl number, $\left(N_{Nuic}/N_{Nuis}\right) - 1(N_{D_2})^{-1/2}$ versus N_{Pr_2}	98
4.3.G	Comparison between experimental and calculated values of laminar Nusselt number (heat transfer to non-Newtonian fluids through coil)	99

NOMENCLATURE

A_j	surface area of the jacketed vessel wall available for heat transfer, cm^2
A_c	outer surface area of the coil tube, cm^2
b_1, b_2, b_3	constants in equations III.30, 33 and 37
b_4, b_5, b_6	
c_1, c_2, c_3	constants in equations III.30, 33 and 37
C_p	specific heat, $\text{cal/gm}^0\text{C}$
D	diameter, cm
D_a	agitator diameter, cm
D_c	diameter of the coil helix, cm
D_t	inner diameter of the straight or coil tube
D_o	outer diameter of the straight or coil tube
D_T	diameter of the agitated vessel
d	differential operator
f	fanning friction factor
f_c	friction factor in coil
f_{sL}	laminar flow friction factor in straight pipe
g	acceleration due to gravity, cm/sec^2
g_c	conversion factor, gm cm/gm(f) sec^2
h_j	heat transfer coefficient for jacketed vessel wall to fluid, $\text{Kcal/hr m}^2\text{ }^0\text{C}$
h_{oc}	coil outside heat transfer coefficient, $\text{kcal/hr m}^2\text{ }^0\text{c}$
h_{ic}	coil inside heat transfer coefficient, $\text{kcal/hr m}^2\text{ }^0\text{c}$
H	height of the fluid level in the vessel, cm
K	consistency index, $\text{gm sec}^{n-2}/\text{cm}$
k'	consistency index, $\text{gm sec}^{n'-2}/\text{cm}$
k	thermal conductivity
n	flow behavior index
n'	generalized flow behavior index
N	speed (r.p.m)
P	pressure, gm(f)/cm^2
P^0	reference pressure, gm(f)/cm^2
ρ	density, gm/cm^3
p	pitch of the helical coil, gm
Q	heat transfer rate, Kcal/min
Q_j	heat transfer rate from the jacketed vessel surface, Kcal/min
Q_c	heat transfer rate from the coil surface, Kcal/min
r	radial distance, cm

t	time, sec
T	bulk temperature of the vessel fluid, °C
T _{ji}	inlet fluid temperature in the jacket, °C
T _{jo}	outlet fluid temperature in the jacket, °C
T _{ci}	inlet fluid temperature in the coil, °C
T _{co}	outlet fluid temperature in the jacket, °C
T _i	inlet temperature
T _s	surface temperature
Δt _j	temperature driving force in the jacket, °C
Δt _c	temperature driving force in the coil, °C
U	average velocity, cm
U _j	jacket overall heat transfer coefficient, Kcal/hr m ²
U _{oc}	coil overall heat transfer coefficient, Kcal/hr m ² °C
u	local velocity in the x direction at r or y, cm/sec
Δx	width of the agitated vessel wall, cm
Y _C	resistance of the jacket side fluid and the Wall

Dimensionless groups

N _{Nu}	Nusselt number, h D/k
N _{Nuj}	Nusselt number, h _j D _T /k
N _{Nuoc}	Nusselt number, h _{oc} D _o /k
N _{Nuic}	Nusselt number, h _{ic} D _i /k
N _{Nuis}	Nusselt number, h _{is} D _i /k
N _{Re}	Reynolds number, D _t Uρ/μ
N _{Re2}	Reynolds number defined by equation 10
N _{Re''}	Reynolds number defined by equation [6]
N _{Rea''}	Reynolds number defined by equation 8
N _{Pra}	Prandtl number defined by equation
N _{pr}	Prandtl number
N _{pr''}	Prandtl number defined by equation VII.20
N _D	Dean number
N _{GZ}	Graetz number, wc _p /kL
N _{prd''}	prandtl number defined by equation [7]

Greek letters

α	constant of proportionality
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δ	boundary layer thickness, cm
θ	function of time, sec
μ	viscosity, gm/cm sec
μ_{ab}	shear or apparent viscosity at bulk temperature
μ_{aw}	shear or apparent viscosity at the wall temperature
μ_2	effective or pseudo shear viscosity, gm/cm
μ_e	effective viscosity
μ''_a	apparent viscosity at the impeller tip
μ_e	effective viscosity
μ''_a	apparent viscosity at the impeller tip defined by equation [13]
μ''_e	effective viscosity at the impeller tip defined by equation [11]
μ_d	differential viscosity, gm/cm sec
τ_w	shear stress, gm(f)/cm ²
τ_s	shear stress, gm(f)/cm ²
c_p	specific heat, cal/gm ⁰ c
$\delta -$	boundary layer thickness, cm

Subscript

a	apparent
a_p	pipe apparent
a_{pw}	pipe apparent at the wall
1,2	inner, outer
b	bulk
d	differential
e	effective
exp	exponential
j	jacket
c	coil
I	inner, inlet
m	mean value
o	outlet, outer
y	yield
SL	laminar flow in straight pipe

1.1 General Introduction

In many if not all branches of industry, one is faced with the problem of designing equipment to transport or to process substances which do not fit any of the classical type of material behavior. Examples of such substances and of processes complicated by their presence are legion.

The mud used in drilling oil wells, is plastic in nature and the paper pulp suspension exhibits unusual time-bounded effects. Spinners of nylon, terylene, etc. into textile fibers encounter non-Newtonian fluids when they transport spinning solutions or force them through spinnerets. Molten plastics show pronounced visco-elastic behavior. Non-Newtonian fluids occur everywhere in the food industry where one deals with pastes, thick suspensions or emulsions. The sanitary engineer encounters anomalous behavior in the transportation of sewage sludge. Many oils and lubricating greases being processed in petroleum industry are non-Newtonian. Visco-elastic and other non-Newtonian fluids are encountered in the pharmaceutical industry in fermentation broths.

Further non-Newtonian problems arise for examples in the flow of blood, in rubber industry, in solid rocket propellants industry, in printing industry, in the development of nuclear reactors using thorium and uranium slurries; and in such applications as the addition of magnesium or boron particles to jet engine fuels for greater thrust, the injection of non-Newtonian fluids in to the boundary layer on immersed bodies such as bearings, ship's hull, and submarines for reduced drag; and the propulsion of rockets by electrostatic acceleration of electrically charged colloidal particles. It is therefore clear that an understanding of non-Newtonian behavior will contribute substantially to the solution of a greater variety of problems.

1.2 Rheological Classification of the Fluids

It is customary to classify real fluids into several categories on the basis of their rheological behavior. A number of classifications are available; however, a universally acceptable one is yet to be evolved. According to the classification presented by Fredrickson (1964) (Appendix B1) the progression is from the least consistent materials (in viscid or Pascalian fluids) which can support no shearing stress whatsoever, to the most consistent materials (Euclidean or rigid solids) which cannot be deformed at all. According to Brodkey (1967) non Newtonian materials can be classified as shear thinning or thickening. In addition, they can be classified according to their time dependency, viscoelasticity and the extent to which they exhibit the effects of the normal stress. There has been much confusion over the classification of non-Newtonian materials, partly because some materials behave in more than one ways depending upon conditions of experiment, degree of concentration, etc. the commonly used classification is that presented by Metzner (1961) (Appendix B2). According to this classification there are four categories of fluids, viz., (i) purely viscous or time dependent fluids, (ii) time dependent fluids, (iii) visco-elastic fluids and (iv) more complex fluids, exhibiting mixed characteristics of more than one of the other categories. Figure 1.2 gives the basic shear diagrams of the various types of fluids.

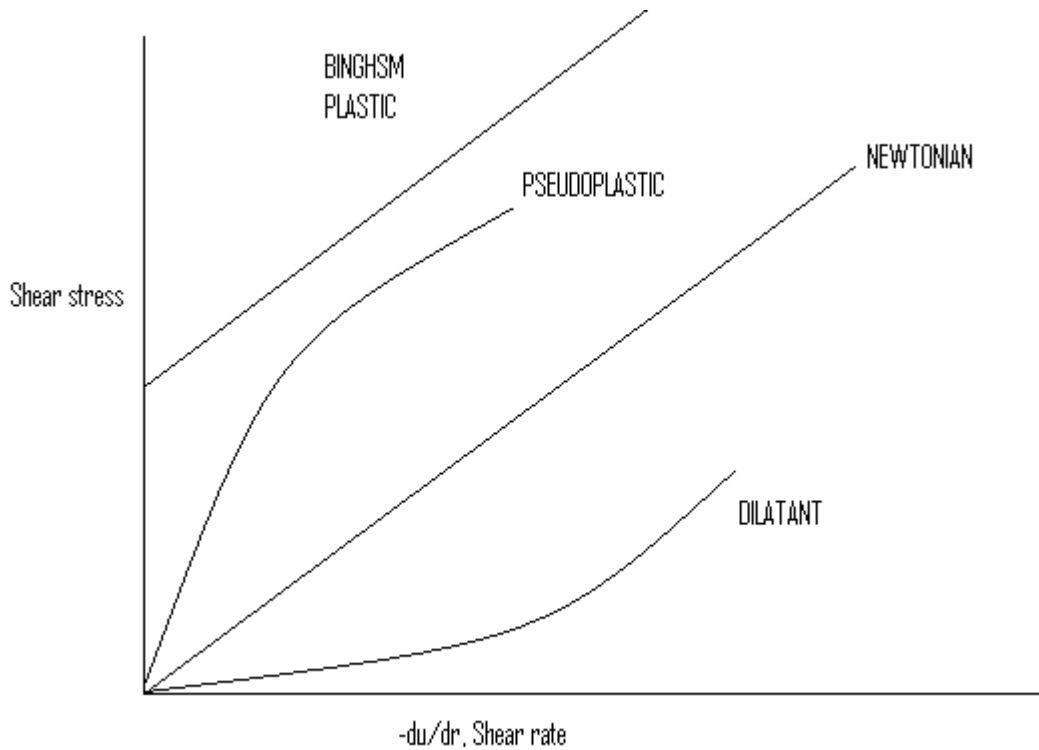


Fig: 1.2: SHEAR STRESS- SHEAR RATE DIAGRAM

Bingham plastic fluid is characterized by a flow curve which is a straight line having an intercept, τ_y , on the shear stress axis. The yield stress, τ_y , is the stress which must be exceeded before flow starts. The equation of the flow curve for stresses above τ_y is

$$\tau - \tau_y = \mu_p \left(\frac{du}{dr} \right), \text{ for } \tau \geq \tau_y \quad (1.2.1)$$

$$\text{And } \left(\frac{du}{dr} \right) = 0, \text{ for } \tau < \tau_y \quad (1.2.2)$$

Where μ_p is called the “coefficient of rigidity” or coefficient of plastic viscosity. Bingham and Green (1919) introduced the above ideal plastic model. Examples of such fluids are: slurries, drilling mud, oil paints, tooth pastes, sewage sludge, suspensions of chalk, grain, rocks, thorium oxide, etc.

After Bingham and Green (1919) introduced the ideal plastic model, Bingham (1922) found that some shear stress thinning materials clearly did not have a yield stress value and were true liquids, even though their viscosity defined by equation (1.2.1) was not constant. Observed flow curves or basic shear diagrams for such materials are shown in figure 1.2. Williamson (1929) introduced the term pseudoplastic to describe these materials. Ostwald (1925) used the expression “a material with structural viscosity” and more recently Reiner (1949) has suggested that materials which possess the rheological properties of viscosity regardless of whether or not it is a constant should be called non-Newtonian liquids. Examples of such fluids are: high polymer solutions such as rubber solution in toluene cellulose ester in organic solvents, emulsions such as GRS rubber latex, detergent slurries, paper pulp suspension, etc. several empirical and semi

empirical models which have been used to describe the behavior of non-Newtonian fluids are summarized in Appendix B3.

Dilatant fluids are similar to pseudoplastic fluids but show rheological behavior opposite to that of pseudoplastic fluids; in that the viscosity increases with increasing shear stress. The power law equation is often applicable but in this case the index is greater than unity. Examples of fluids which show dilatancy are pigment-vehicle suspensions such as paints and printing inks when the solid content is high, starch, potassium silicate, quick sand and beach sand in water and vinyl resin plastic solutions have been reported to show pseudoplastic behavior at low deformation rates and dilatant behavior at high deformation rates. Ammonium oleate suspensions show the increase in viscosity with shear rate even though they do not dilate. Hence, Brodkey (1967) suggests that these materials should be referred to simply as shear thickening materials.

Apparent viscosity of more complex fluid depends not only on the rate of shear but also on the time for which the shearing force has been applied. Fluids having properties which are time dependent are classified into two groups, depending on whether the shear stress at a constant deformation rate decreases or increases with time. These are known, respectively, as thixotropic and rheopectic fluids.

With purely viscous fluids, energy imparted to the system as work during shear is dissipated as heat. In contrast, the energy imparted to an ideally elastic material during strains is stored as potential energy and may be recovered upon removal of the stress. Viscoelastic fluids distribute imparted energy according to both the mechanisms and may be defined as the fluids having both viscous and elastic properties.

Viscoelasticity, characteristic of long chain molecules, is fostered by elastic elements linked together by the frictional restraints of molecular entanglements and strong inter molecular forces. Polymer chains, that possesses coiled configuration at conditions of rest, are placed under tension and extended when subjected to shear, thereby store potential energy. The strains caused by the extensions of these elastic elements becomes superimposed upon the motion of viscous shear. Polyethylene melts and napalm and similar jellies, polymer melts, etc., offer typical examples of viscoelasticity.

Frequent occurrence of such fluids in industries has created lot of interest in the study of the behavior of these fluids in various process equipment. In many cases such fluids flow through curved pipe such as helical or spiral coils for the exchange of heat to or from the fluids. In reaction vessels such fluids are agitated and heat removal takes place by cooling coils or through jacket arrangements. For the design of equipment involving such systems it is extremely important to investigate the heat and momentum transfer behavior of non-Newtonian fluids in coils and agitated vessels.

1.3 Heat Transfer in Stirred vessels

In many industrial processes it is frequently necessary or convenient to perform one or more operations in a mechanically stirred vessel. Among these are blending of two miscible fluids,

dispersion of one phase into other immiscible one, suspension of solid particles into a liquid, heating and cooling of fluids or a dispersion of fluid or solid into another fluid, gas absorption in a liquid or liquid (or solid) – liquid dispersion, transfer of mass from one phase to another and the conduct of chemical reactions.

All these stirred vessel, vessel operation, whether done continuously or in a batch fashion have several things in common. They consume certain amount of power. There is nearly always a preferred geometry of the vessel-agitator combination for any specific operation. All of these processes are rate processes and they take a finite time for their completion. The literature contains fairly good reports of work on individual operations of blending, heat transfer, etc., but their interactions on one another have been studied only slightly.

The flow of heat or out of a stirred vessel is an operation of great practical importance. By far the most popular method for heat transfer in either direction is forced convection. The forced convection is commonly done by one of the varieties of impeller-vessel geometries.

In processes where there exists gradients in transport properties in the bulk and which are detrimental to the process system, the mechanical device is often sought after to reduce the gradients and save the process system. The mechanical device brings an ordered macroscopic homogeneity, by continuously intermingling the species of the system in the bulk phase. In the case of heat transfer, where temperature differences exist in bulk fluid between an equipment surface and the fluid, or between suspended particles and the continuous phase, the mechanical device, capable of transferring materials to and from the equipment surface or the particles, reduces the thickness of the transport resisting film or in other terms increases the temperature gradient immediately adjacent to the equipment surface and the bulk fluid.

When the mechanical devices are used to provide mixing action to the bulk of fluid, the fluid inertia exerts a resisting force to the change in direction of motion or its velocity. Viscous drag or fluid shear forces provide another type of resistance to fluid motion. For low viscosity fluids like water in which turbulence is readily induced, the inertia of the fluid provides not only the major resistance to stirring the fluid but also the major method by which fluid movement is transmitted to parts of the fluid which are remote from the stirrer. For viscous materials like polymers and polymer solutions, shear forces are not only the major resistance to the motion of the stirrer, but also provide a mechanism for moving the fluid in a desired flow pattern.

For mixing, both the exterior and the interior type of motive agencies are in use. However, for most practical purposes, the interior motive agencies like agitators for mixing are best to use. For heat transfer in agitated vessel the agitators are categorized as proximity when the blades sweep close to or scrap the vessel wall and as non-proximity when the agitator blades rotate with considerable clearance from the wall of the vessel. The non-proximity agitators are used for mobile to medium viscous (0.1 to 1000cp) fluids. These smaller generally higher speed agitators also facilitate heat transfer to the surfaces or coils immersed in the vessel as well as in the jacket vessel walls. Such agitators are propellers, turbines and paddles. The proximity type agitators are used for highly viscous fluids, non-Newtonian, semi-solid suspensions or granular solid materials. The examples of these types of agitators are anchor type, or wall scraping type.

Because of the extensive use in industries, the systems of heat exchange in a jacketed agitated vessel with non-proximity agitators have received considerable attention in the recent years. Unfortunately, there is large divergence in the correlation presented by the previous workers. The reasons of this fact are mainly the difficulties in obtaining the velocity and temperature profiles for the proper mathematical analysis. The choice of viscosity in dimensionless groups (Reynolds and Prandtl numbers) has created another problem in obtaining a suitable correlation for non-Newtonian fluids. From the available literature in this field, it is clear that the correlations presented for the non-Newtonian fluids are based upon Newtonian considerations. Reynolds and Prandtl number indices are similar in both the cases, but with different geometrical constants. The viscosity terms taken in Reynolds and prandtl numbers have been taken equivalent to that of fluid flowing through a pipe rather than that in a mixing vessel. The dimensionless groups defined by most of the earlier workers do not truly predict the flow behavior in an agitated vessel. Thus more work is needed to obtain a single valued heat transfer correlation for both Newtonian and non-Newtonian fluids in agitated vessel and coil assemblies.

1.4 OBJECTIVE

From the available literature review in chapter 2 it is seen that the average heat transfer coefficient depends on the geometrical and physical factors. The correlation between them is not so simple because of the complex relationship between the large numbers of variables involved. For example the heat transfer rate in an agitated vessel, either from the vessel wall or from the wall of coils, depends on velocity and temperature profiles near the wall. The velocity profile itself, which affects the temperature profile, depends on geometrical factors, such as the impeller diameter and its nature, position and speed; the diameter of the tank, the diameter and pitch of

the coils, the diameter of the coil cube, the fluid height in the tank and also on the position and the dimensions of the baffles. The physical parameters are the physical properties of the fluid, specially, the viscosity, the thermal conductivity, density, specific heat and their variation with the temperature. Physical parameters are normally grouped in terms of Prandtl number, Reynolds number and viscosity correlation factors where as geometrical parameters are considered in the Reynolds number and various geometric dimensionless groups.

The non-linear relationship between shear stress and shear rate for non-Newtonian materials make the developments of heat transfer correlation more complex. For such materials difficulties arise in specifying the Reynolds number, Prandtl number and viscosity ratio. Therefore an appropriate choice of viscosity is necessary to obtain a suitable correlation for heat transfer. There are several objectives in this part of the present investigation. These are

1. To obtain heat transfer coefficients in jacketed mixing vessel for various impeller diameter and speed for both Newtonian and time dependent power law type non-Newtonian fluids.
2. To obtain heat transfer coefficients from helical coils to agitated Newtonian and non-Newtonian fluids.
3. To obtain laminar flow heat transfer coefficients inside the coiled tube at various flow rates for both Newtonian and non-Newtonian fluids.

All these coefficients will be determined under the steady state condition and the data correlation will be based on the following theoretical background.

CHAPTER 2

REVIEW OF LITERATURE

In the following pages the available literature in the field of heat transfer to Newtonian and non-Newtonian fluids flowing through coils and in jacketed mixing vessels has been discussed.

D. Pierce and Phillip (1924) discussed the use of agitation as a means of increasing the rate of heating or cooling in places where it is difficult to provide more transfer surface. They carried out laboratory experiments and obtained the effect of agitator speed upon transfer coefficient by cooling the warm contained in a 300 gallon, 4 ft diameter and 3 ft deep tank by means of water passing through a 121 ft long and $\frac{3}{4}$ inch i.d. lead pipe coil wrapped around the inside of tank. An agitator with 39.5 inch long blade was used for agitating the contents of the tank. A marked effect of agitation on heat transfer coefficient was found. The plots of overall heat transfer coefficients versus agitator speed indicated a continuous but not proportionate improvement in the overall coefficient as the speed of agitation increased. But the plot of external surface coefficient versus agitator speed gave a straight line without showing any point of inflection. Authors also plotted various resistances in question versus agitator speed and showed a pronounced decrease in total resistance and external surface resistance with increase in agitator speed from 0 to 40 rpm. The inner surface resistance of the coil remained practically the same.

Sieder and Tate (1936) presented the experimental data on heat transfer to three coils of widely varying dc/Dt ratios. They included the viscosity ratio term in their correlation and obtained 0.14 as the exponent for the same. Finally, the authors developed a correlation:

$$\frac{hD_t}{k} \left(\frac{c_p \mu}{k} \right)^{-1/3} \left(\frac{L}{D_t} \right)^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{-0.14} = 1.86 \left(\frac{D_t G}{\mu_b} \right)^{1/3} \quad (2.1)$$

Hixon and Baum (1941) studied the heat transfer in liquid-solid agitated systems to show that an analogy exists between heat and mass transfer. The equation was correlating heat transfer data for several liquids; water, benzene, acetic acid and nitro-benzene, in a series of geometrically similar vessels using a series of dimensionally similar turbine agitators, was presented. Each liquid and its frozen solid pieces of known area and weight were agitated in properly insulated vessels of several sizes. Heat transfer coefficients, h were calculated for each run. Several runs for each liquid were then taken by changing the agitator speed. A correlation for all the points with an average deviation of less than $\pm 10\%$ was proposed as

$$\frac{hD_T}{k} = 0.207 \left(D_a^2 N \rho / \mu \right)^{0.63} \left(\frac{c_p \mu}{k} \right)^{0.5} \quad (2.2)$$

The above correlation includes the data for fluids having a threefold range in the viscosity, a four-fold range in the thermal conductivity and a three-fold range in the specific heat. The size of vessels, D_T varied between 15.2 cm to 119 cm and the corresponding range in volumes by five hundred. The speed range was 160-450 rpm. Reynolds number and Prandtl number varied from 1.53×10^4 to 1.22×10^6 and from 7.1 to 23.8, respectively.

Chilton, Drew and Jebens (1944) carried out several experiments to study the steady state heat transfer as well as the batch heat transfer in a jacketed, 1ft diameter, steel, dish-bottomed vessel. Inside the vessel a coil of ½ inch i.d. copper tubing, 5.75 inch height and of 2.42 sq ft outside area, cylinder was installed for the flow of cooling water. A flat peddle 0.6 ft long was used for agitation. Agitator speed ranged from 5.0 to 1000 rpm and the corresponding Reynolds number from 300 to 4×10^5 . The system under study was water, 92% glycerol, light medium lubricating oil (A 12 oil) and heat transfer oil (LM oil). The temperature of the liquids in the vessel varied by 1-3 °C from point to point but this variation decreased at higher agitator speeds. Authors presented following correlations for jacket and coil, respectively. For jacket

$$\frac{h_j D_T}{k} = 0.36 \left(D_a^2 N \rho / \mu \right)^{\frac{2}{3}} \left(\frac{c_p \mu}{k} \right)^{\frac{1}{3}} \left(\frac{\mu}{\mu_j} \right)^{0.14} \quad (2.3)$$

And for coil

$$\frac{h_c D_T}{k} = 0.87 \left(D_a^2 N \rho / \mu \right)^{\frac{2}{3}} \left(\frac{c_p \mu}{k} \right)^{\frac{1}{3}} \left(\frac{\mu}{\mu_j} \right)^{0.14} \quad (2.4)$$

Reynolds and Prandtl numbers under study ranged from 286 to 6.62×10^5 and from 2.67 to 1060 respectively.

Brown et al. (1947) obtained the heat transfer data in 750 gallon, cast iron sulphonators and nitrators of 5 ft id. with hemispherical bottoms. Anchor type agitators were used in two sulphonators and four bladed (45° pitched downward thrust) propeller agitator of 2 ft diameter was used in nitrator. Both batch and steady state heat transfer data were taken. Above a water flow rate of 4000 gph, authors observed that the water film coefficient, h_w , is directly proportional to $V^{0.87}$, V being the cooling water rate in gph. The following correlation was proposed:

$$\frac{hD_T}{k} = 0.55 (N_{Re})^{\frac{2}{3}} (N_{Pr})^{0.25} \left(\frac{\mu}{\mu_j}\right)^{0.14} \quad (2.5)$$

Cummings and West (1950) made an study of forced convection heat transfer in an agitated, jacketed, 100 gallon, 30 inch diameter, stainless steel kettle of dished bottom. The data were taken with six liquids; water, toluene, isopropyl alcohol, ethylene glycol, glycerol and white mineral oil. A 2 inch diameter helical coil constructed of 10 turns of 1 inch o.d., stainless steel tubing were installed inside the vessel for circulating cooling water. Agitation was supplied by a retreating 6 blade turbine impeller of 12 inch diameter placed at a height of 10 inch from the bottom of the kettle. In a way the work may be considered as an extension of the work reported by Chilton et al. (12) to include film coefficients measured in large equipment which is geometrically similar to the type ordinarily encountered in industrial practice. In all they employed 3 different impellers, pitched blade turbines, and curved blade turbines within the speeds ranging from 107 to 240 rpm corresponding to Reynolds number values of 2X10³ to 7X10⁵. Most inconsistent results were obtained with water. The film coefficients increased with increase in the speed of agitation. The presence of solids in the batch tended to lower the film coefficients. The correlations obtained by the authors for both heating and cooling the fluids in the vessel are: for jacket

$$\frac{h_j D_T}{k} = 0.40 \left(D_a^2 N \rho / \mu \right)^{\frac{2}{3}} \left(\frac{C_p \mu}{k} \right)^{\frac{1}{3}} \left(\frac{\mu}{\mu_j} \right)^{0.14} \quad (2.6)$$

And for coil

$$\frac{h_c D_T}{k} = 1.01 \left(D_a^2 N \rho / \mu \right)^{0.62} \left(\frac{C_p \mu}{k} \right)^{\frac{1}{3}} \left(\frac{\mu}{\mu_c} \right)^{0.14} \quad (2.7)$$

Inconsistent results obtained with water in the kettle for individual film coefficients under various conditions could perhaps be expected since small amount of dirt or scale on the coil surface or the presence of air in the batch during agitation would tend to affect the coefficients to a large extent. Variation in the water film coefficients on the outside of the coil may be due to the minor resistances which cause experimental errors to a much greater extent than the outer systems. Higher agitation speeds generally produced higher coefficients except in case of water. A satisfactory explanation for this discrepancy could not be easily found out.

The correlation proposed by this workers were found to yield film coefficients about 16% higher for the coil and about 11% higher for the jacket than the equations presented by **Chilton et al. (1944)**.

Oldshue and Green (1954) have investigated helical coil heat transfer in baffled and agitated vessels using water and turbine oil (0.4 to 400 c.p.) as the test fluids. They used disc-flat bladed turbine impellers to various diameters ranging from 12 to 28 inch. A special feature of their coil was that it had the combination of alternate coil turns for heating and cooling. They studied the speed range of 52 to 230 rpm corresponding to wide range of Reynolds number ranging from 400 to 1.5×10^5 . In their investigation, which was carried out under steady state, the outside heat transfer coefficients were interpolated from the measured surface film transfer coefficients of the heating and cooling coils at a viscosity ratio of 1.0. They have proposed the following correlation:

$$N_{Nu} = 0.17 N_{Re}^{0.67} N_{Pr}^{0.37} \left(\frac{D_a}{D_T}\right)^{0.1} \left(\frac{D_t}{D_T}\right)^{0.5} \quad (2.8)$$

An interesting point of their investigation is that they reported 0.37 as the exponent for prandtl number as against the common value of 0.33. The power for the viscosity factor in this investigation was 0.18 at $\mu = 1000$ c.p. and 0.93 at $\mu = 0.3$ c.p. (these are, of course the extrapolated values). The higher figure is unlikely and is probably due to strong natural convection effects during tests with water. According to **Brown et al. (13)** when hoc is plotted against the slope would work out to be greater than -0.409. Surprisingly enough the results of **Oldshue et al.** follow this correction rather well.

Metzner and Otto (1957) studied the agitation or mixing of fluids for the development of a general relationship between impeller speed and shear rate which was used to correlate the power consumption data of non-Newtonian fluids by the use of a generalized form of the conventional power number-Reynolds number plots for Newtonians. They used flat-bladed turbines of 2 to 8 inch diameter. Tank diameter ranged from 6 to 22 inch and power input from 0.5 to 176 h.p. per 1000 gallons. The authors suggested that the shear rate around an impeller is directly proportional to the impeller speed and the quantitative relationships which are applicable to both Newtonian and non-Newtonian fluids represent a simple generalization of the well-known results for Newtonian systems, i.e. Ruston and Oldshue equation

$$N_p = 71/N_{Re} \quad (2.9)$$

Is the same for Newtonian and non-Newtonian fluids and the equation

$$P = 71 \frac{\mu_a}{g_c} D_T^3 N^2 \quad (2.10)$$

Is particularly useful for the design purposes of mixing systems. They further concluded that more power is required for rapid mixing of highly non-Newtonian systems than for Newtonian fluids and laminar region may extend to higher Reynolds number in pseudoplastic fluids than in Newtonian systems.

Gary and Su (1960) studied heat transfer in unbaffled and baffled, jacketed, dish bottomed, 20 inch diameter, 30 gallon stainless steel vessels with a mild steel jacket using a six flat bladed turbine impeller of 6 inch diameter. Available surface area for heat transfer was 1.5 sq. ft.

In correlating the data, authors assumed no variation in U_j over the temperature subrange, in specific heat, in rate of agitator work and no scale build up. Radiation losses and conversion of mechanical energy to sensible heat and change in thermal conductivity were assumed to be negligible. No significance differences were obtained among one, two or four baffle conditions. Baffling increased the kettle-side film heat transfer coefficient, h_j by 37%.

The data taken in viscous region and in transition region showed no difference between baffled and unbaffled conditions. The difference existed in turbulent region only. The purpose correlations are: for baffled vessel

$$\frac{h_j D_T}{k} = 0.54 \left(D_a^2 N \rho / \mu \right)^{\frac{2}{3}} \left(\frac{C_p \mu}{k} \right)^{\frac{1}{3}} \left(\frac{\mu}{\mu_j} \right)^{0.14} \quad (2.11)$$

And for unbaffled vessel

$$\frac{h_j D_T}{k} = 0.74 \left(D_a^2 N \rho / \mu \right)^{\frac{2}{3}} \left(\frac{C_p \mu}{k} \right)^{\frac{1}{3}} \left(\frac{\mu}{\mu_j} \right)^{0.14} \quad (2.12)$$

Ackley (1960) investigated coil side and jacket side heat transfer, in agitated vessels of 30-10000 gallon capacities using water as the test fluid and square pitch propellers and three bladed turbine agitators. He proposed the correlation

$$N_{Nu} = 0.078 N_{Re}^{0.62} N_{Pr}^{0.33} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (2.13)$$

Strek (1963) investigated the effect of variation in system geometry on heat transfer using two flat bottomed vessels of 11.83 and 23.6 inch diameters equipped with 6 flat bladed turbine agitators. The Reynolds number varied from 5×10^4 to 8.5×10^5 in the vessel which had four equally spaced, flat-walled baffles, each of width 1/10 of the vessel diameter. He presented following correlation assuming the exponents of N_{Re} and N_{vis} from Chilton et al. (1944)

$$N_{Nu} = 1.01 (N_{Re})^{0.66} (N_{Pr})^{0.33} (N_{vis})^{0.14} \left(\frac{H}{D_T}\right)^{0.12} \left(\frac{D_a}{D_T}\right)^{0.13} \quad (2.14)$$

Chapman et al. (1964) obtained heat transfer data in baffled, 12 inch diameter, jacketed, mild steel, flat bottomed vessel and also in 15, 18 and 27 inch diameter flat bottomed stainless steel vessels. Heating due to mechanical energy of rotating impellers was found to be negligible. The correlation proposed by the authors for the data obtained in all the four tanks is

$$N_{Nu} = 1.15 (N_{Re})^{0.65} (N_{Pr})^{0.33} \left(\frac{\mu}{\mu_j}\right)^{0.24} \left(\frac{H}{D_T}\right)^{0.4} \left(\frac{Z}{D_T}\right)^{0.56} \quad (2.15)$$

And covered a Reynolds number range of 20 to 4×10^4 .

Beckner and Smith (1966) investigated the effect of clearance between the anchor and the vessel wall, of breaker bars, and of using flat and pitched bladed anchors on power input for Newtonian and non-Newtonian fluids agitated with anchor-shaped impellers. A generalized correlation for power input was derived as

$$\left(\frac{P}{N^3 D_T^5 \rho}\right) \left(\frac{C}{D_T}\right)^{\frac{1}{4}} = 82 \left[\frac{N^{2-n} - D_a^2 \rho}{K\{a(1-n)\}^{n-1}}\right]^{-0.93} \quad (2.16)$$

Where a is the geometric parameter defined by the equation

$$K = a(1-n). \quad (2.17)$$

Jha and Raja Rao (1966) undertook a very systematic investigation of heat transfer in agitated vessels. They studied the variation of heat transfer in vessels with and without baffles and also with a variety of test fluids such as water, glycerol and suspensions of silica sand, aluminum and iron powder in water. They took measurements in a flat bottomed cylindrical vessel together with a copper helical coil of 1/2 inch diameter tubing. They used 4 inch diameter impeller for the agitation of test fluids. For the Reynolds number range of 1×10^5 to 2.5×10^5 they proposed the correlation:

$$N_{Nu} = 1.04 N_{Re}^{0.67} N_{Pr}^{0.33} \left(\frac{D_T H}{T}\right)^{0.18} \left(\frac{B}{6}\right)^{0.28} \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (2.18)$$

Which seem to obey all the accepted indices.

Carreau et al. (1966) studied batch heat transmission to agitated non-Newtonian pseudoplastic liquids in a jacketed, 80 gallon, 30 inch diameter, stainless steel vessel with four bladed, 40° pitched turbine agitator. Heating or cooling of the vessel contents were done by passing steam or water in the jacket. Measurements were made in the Reynolds number range of 100 to 5000, Prandtl number range of 100 to 800 with fluids having flow behavior indices of 0.343 to 0.633. Energy dissipation (heat due to agitation) and heat losses were shown to be negligible compared to total flux and tended to compensate one another. Liquid film coefficient h_j was calculated from the overall coefficient U_j using a modified Wilson plot. Generalized Reynold's number, N'_{Re} was used in determining its effect on Nusselt number. An average value of 0.66 was obtained as the index of N'_{Re} . An attempt to use generalized Prandtl number N'_{Pr} to determine its influence on heating or cooling was unsuccessful. The use of differential viscosity at high shear rates gave the Prandtl number index to be 0.265 for heating and 0.261 for cooling. Two equations, correlating the data for heating and cooling were given as

$$N_{Nu} = 3.41 (N'_{Re})^{\frac{2}{3}} (N_{Pr})^{1/3} \quad (2.19)$$

$$N_{Nu} = 1.43 (N'_{Re})^{\frac{2}{3}} (N_{Pr})^{1/3} \quad (2.20)$$

Final correlation combining heating and cooling both was represented by the equation

$$\frac{h_j D_T}{k} = 1.474 \left[\left(\frac{D_a^2 N^{2-n} \beta}{K} \right) 8 \left(\frac{n}{6n+2} \right)^n \right]^{0.70} X \left(\frac{C_p \mu_{d\infty}}{k} \right)^{0.33} \left(\frac{\mu_{dw}}{\mu_d} \right)^{-0.24/n} \quad (2.21)$$

$$\text{For } \frac{D_a}{D_T} = \frac{3}{5}, 0.34 \leq n \leq 0.63, 100 \leq N_{Re} \leq 5000$$

Gluz and Pevlushenko (1966) like **Pollard and Kantyka (1969)** suggested that the correlation developed from Newtonian systems could be used to predict the non-Newtonian data if μ_A is replaced by μ . They correlated the experimental data by Equations

$$\frac{h_j D_T}{k} = 0.215 N_{Re}^{0.67} N_{Pr}^{0.33} \left(\frac{K_w}{K}\right)^{0.18} \quad (2.22)$$

For turbine agitator and

$$\frac{h_j D_T}{k} = 0.636 N_{Re}^{0.67} N_{Pr}^{0.33} \left(\frac{K_w}{K} \right)^{0.18} \quad (2.23)$$

$$8 < N_{Re} < 30$$

$$\frac{h_j D_T}{k} = 0.371 N_{Re}^{0.67} N_{Pr}^{0.33} \left(\frac{K_w}{K} \right)^{0.18} \quad (2.24)$$

$$30 < N_{Re} < 3 \times 10^5$$

For the anchor agitator.

Skelland and Dimmick (1969) obtained data on coil heat transfer to develop a generalized correlation applicable to all time independent non-Newtonian fluids, thereby eliminating restrictions associated with particular empirical relationships for fluids such as Bingham plastics and Ellies or power law fluids. They conducted the steady state flow tests with 0.1 and 0.3% of carbopol-934 solutions and the data were taken for heating and cooling with both upthrusting and downthrusting agitation for substantial ranges of impeller speed, equipment geometry and the fluid physical properties. The densities, specific heat and thermal conductivities of the fluids were taken to be the same as those for water. The authors proposed the following general correlation applicable to all time-independent non-Newtonian fluids:

$$\frac{h_c D_o}{k} = 0.258 \left(\frac{D_a^2 N}{\mu_a} \right)^{0.62} \left(\frac{C_p \mu_a}{k} \right)^{0.32} \left(\frac{\mu_a}{\mu_{aw}} \right)^{0.2} \left(\frac{D_a}{D_T} \right)^{0.1} \left(\frac{D_o}{D_T} \right)^{0.5} \quad (2.25)$$

And for special cases of power law fluids a separate equation was given. Results covered the Reynolds number range of 332 to 2.6×10^5 , Brandt number of 11.8 to 1110.3 and n of 0.528 to 0.91.

Sandal and Patel (1969) reported experimental results on batch heat transfer to pseudoplastic fluids (aqueous carbopol solution) in a 7.218 inch diameter, jacketed agitated vessel. Agitation was supplied by six-bladed flat blade disc turbine of 2.5 inch diameter and 7.1 inch diameter anchor agitator. The data were correlated with a Sieder-Tate type equation based on the use of an effective viscosity in the expression for the generalized Reynolds and Brandt numbers. The effective viscosity μ_e was determined by the application of the relationship between impeller speed and the average shear rate existing in the vessel as proposed by the previous investigators. For the turbine impeller the data extended over the Reynolds numbers from 80-93,000 and Brandt numbers from 2.1 to 644. Data for anchor agitator were obtained in the Reynolds number range of 300 to 90,000 and a Brandt number range of 2.1 to 644. The authors recommended the

following correlations with average deviations of $\pm 11.3\%$ and $\pm 18.3\%$, respectively. For anchor agitator

$$N_{Nu} = 0.315 (N_{Re})^{\frac{2}{3}} (N_{Pr})^{1/3} \left(\frac{\mu}{\mu_j}\right)^{0.12} \quad (2.26)$$

And for turbine agitator

$$N_{Nu} = 0.482 (N_{Re})^{\frac{2}{3}} (N_{Pr})^{1/3} \left(\frac{\mu}{\mu_j}\right)^{0.12} \quad (2.27)$$

Coyle et al. (1970) carried out an extensive study of heat transfer in jacketed mixing vessels with close clearance impellers using fluids of viscosities over 10,000 centipoises, which is the highest viscosity reported so far in the literature. Two different vessels; one of 14 inch diameter with a 13 inch diameter impeller and the other of 30 inch diameter with 29 inch diameter impeller, were used to examine the heat transfer coefficients with helical, anchor and scraper blade agitators in a wide variety of organic materials of different viscosities and pseudo plastic characteristics. An interesting result of their investigation was that the heat transfer coefficients with helical impellers were related to the clearance between the impeller and the tank wall and were not influenced, to any important degree, by the speed or the fluid viscosity. The mechanism of viscous mixing was approximated by the rate of conduction through a stagnant film of approximately half the clearance and ‘convection type’ correlation was said to be useful in describing the effect of operating variables. The authors expressed the possibility of obtaining a higher coefficient in large equipment due to thinner film formation at a constant impeller to tank diameter clearance.

Rajshekhran, Kubair and kuloor (1970) reported heat transfer data for non-Newtonian fluids flowing through spiral and helical coils. They studied 0.5, 1.0, 1.5% CMC, 1% sodium silicate and 0.5% CPM solutions in spiral and helical coils of curvature ratios ranging from 0.031 to 0.056 on the same apparatus as that reported by Kubair and Kuloor (K10) for Newtonian fluids. Within the Graetz number range of 20 to 103, they proposed the correlation:

$$N_{Nu} = 1.98 + 1.8 \left(\frac{D_t}{D_c}\right) \left(\frac{3n+1}{4n}\right)^{0.7n} N_{Gz}^{0.7} \quad (2.28)$$

Akiyama and Cheng (1971) attempted a finite difference solution using line iterative and boundary vorticity method for the hydrodynamically and thermally fully developed laminar forced convection in curved pipe subjected to the thermal boundary conditions of axially uniform wall heat flux and peripherally uniform wall temperature at any axial position. They have shown that all the heat transfer results for N_{pr} greater than or equal to 1.0, can be correlated by a single curve using a new parameter ($K^2 N_{pr}$) with reasonable accuracy. The following approximate equation has been deduced using the new parameter $K^2 N_{pr}$ as the curve to best fit all the numerical results:

$$\left(\frac{N_{Nu}}{N_{Nu0}}\right) = 0.181 Q(1 - 0.839 Q^{-1} + 35.4 Q^{-2} - 203 Q^{-3} + 419 Q^{-4}) \quad (2.29)$$

Where $Q = (K^2 N_{pr})^{\frac{1}{4}} \geq 3.5$ or $N_{pr} \geq 1.0$

For $Q \leq 3.5$ the secondary flow is not considered to be important in that region.

The above authors recalculated heat transfer results from the above study using the parameter $K N_{pr}^{1/2}$ instead of $K^2 N_{pr}$. The new correlation has been found to be very effective since all the results for $N_{pr} \geq 1$ nearly coincide. The correlating equation for Nusselt number is

$$\left(\frac{N_{Nu}}{N_{Nu0}}\right) = 0.270 Q(1 - 1.48 Q^{-1} + 23.2 Q^{-2} - 120 Q^{-3} - 212 Q^{-4}) \quad (2.30)$$

Where $Q = (K N_{pr}^{1/2})^{\frac{1}{2}} \geq 3.0$ for $N_{pr} \geq 1.0$ which is true for all the laminar flow regimes and N_{Nu0} is 3.66 for uniform wall temperature and 4.36 for the uniform heat flux.

Martone and Sandall (1971) characterized their test liquid as Bingham plastic for which

$$\tau = \tau_y + \mu_p \gamma \quad (2.31)$$

And gave the following correlations. For turbine blades

$$\frac{h_j D_T}{k} = 0.534 N_{Re}^{2/3} N_{Pr}^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \left(\frac{\phi}{1-\phi}\right)^{0.065} \quad (2.32)$$

And for anchor agitator the equation is:

$$\frac{h_j D_T}{k} = 0.315 N_{Re}^{2/3} N_{Pr}^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.2} \left(\frac{\phi}{1-\phi}\right)^{0.072} \quad (2.33)$$

Heinlein and Sandall (1972) extended the work of **Martone and Sandall (1971)** using the same experimental apparatus with major modification being that Chromel-Alumel thermocouples were used to measure the temperatures instead of iron-constantan as used in the previous study. Both heating and cooling of the vessel fluid was studied by introducing either steam or cooling water into the vessel jacket. Pseudoplastic (carbopol solutions) and Bingham plastic (chalk slurries) fluids were used in the present study whose rheological properties were measured by a capillary tube viscometer. Within the Reynolds number range of 12-300, the authors correlated their data by the equation:

$$N_{Nu} = C N_{Re}^{1/2} N_{Pr}^{1/3} N_{visc}^{0.18} \quad (2.34)$$

Suryanarayanan, Mujawar and Raja Rao (1973) investigated the coil-outside and vessel-inside film heat transfer coefficients for dilute polymer solutions, for standard and non-standard configuration of jacketed agitated vessel. The polymer solutions in the vessel were heated by hot water flowing in the jacket. Its temperature was maintained constant at any desired value by the flow cold water in the helical coil. The speed of agitation varied from 180 to 670 rpm. The authors proposed the following correlation: for coil side

$$N_{Nuc} = 0.236 (N'_{Re})^{0.66} (N'_{Pr})^{0.33} \left(\frac{D_a}{D_T}\right)^{0.17} \quad (2.35)$$

And for jacket-side

$$N_{Nu_j} = 0.254 (N'_{Re})^{0.63} (N'_{Pr})^{0.33} \left(\frac{D_a}{D_T}\right)^{0.14} \quad (2.36)$$

Mishra et al. (1974) presented experimental results for agitated pseudo plastic liquids in a jacketed, hemispherical bottomed, 1 ft diameter stainless steel vessel using three bladed propeller type impeller. A new type of effective viscosity expression adequately describing the flow behavior was used in the Reynolds and Prandtl groups. Based upon the data taken with six aqueous CMC solutions, the following correlation:

$$N_{Nu} = 1.84 (N''_{Re})^{0.6} (N''_{Pr})^{0.33} \left(\frac{\mu_e}{\mu_{we}}\right)^{0.14} \quad (2.37)$$

Covering the modified Reynolds number (N''_{Re}) range of 7×10^2 to 2×10^5 and modified Prandtl number (N''_{Pr}) range of 25 to 750, was proposed.

S. D. Joshi , A. E. Bergles(1980) An experimental study is reported of heat transfer to laminar flow of two water-methocel pseudoplastic (power law) solutions in a circular tube subjected to a uniform wall heat flux. The object of this study was to evaluate the effects of non-Newtonian behavior and temperature-dependent consistency on heat transfer. The experimental Nusselt numbers are compared with numerical predictions and experimental data. Two correlations are recommended according to the temperature-dependence of the rheological characteristics.

Strek and Karcz [1988] studied the local heat transfer using one and two standard Rushton turbine-agitated gas-liquid systems in a 0.3 m diameter baffled vessel. The fluids investigated were distilled water, an aqueous solution of Na_2SO_4 , and glycerol, each with air. The measurements of the local heat transfer coefficients at the wall of the stirred tank were carried out by means of a heat flux meter. The local values of the heat transfer coefficient h_j were calculated on the basis of directly measured local values of the heat flux per unit area Q_r and the temperature difference A_t between the wall and the bulk liquid, $h_j = Q_r/A_t$. The graphical result suggests that the greatest values of the heat transfer coefficient at the vessel wall occur in the plane of the impellers. This has been demonstrated widely [1936, 1951-1954] for single-phase systems, and more recently it was found to apply to two-phase systems as well [1959]. However, for the two-phase system the heat transfer and hydrodynamics are more complex [1959]. Thus the prediction by Strek and Karcz [1951] for two-phase systems can be applied only in the hydrodynamically completely dispersed zone, as shown in Fig. 2c. Furthermore, a significant difference is observed between dispersion in pure water and that in salt solution. In the water-air system the bubbles tend to coalesce, whereas in the salt water-air system, depending on the type and concentration of salts, the coalescence of the primarily produced gas bubbles is more or less hindered, and the local heat transfer is slightly increased with increases in salt concentration. The result suggests that coalescence bubbles cause a decrease in heat transfer; this is possibly due to the decreased turbulence associated with the reduction in total bubble surface area that occurs with coalescence.

Enio Kumpinskyt (1995) studied Transient models for sensible heat were developed to assess the thermal performance of agitated vessels with coils and jackets. Performance is quantified with the computation of heat-transfer coefficients by introducing vessel heating and cooling data into model equations. Of the two model categories studied, differential and macroscopic, the latter is preferred due to mathematical simplicity and lower sensitivity to experimental data variability.

It is possible to calculate individual heat-transfer coefficients if each heat-transfer unit is equipped with a flow meter and temperature sensors, upstream and downstream of the reactor. It is not uncommon, however, to have unknown service fluid flow rates in an industrial setting. The macroscopic balances may be used to calculate overall heat-transfer coefficients even under this circumstance. The methods of this work can be used to assess chemical reactor cooling capabilities at peak exotherm, in combination with laboratory calorimetric data. Finally, a simple equation was written that can be employed to do quick heat transfer calculations in the field, providing reasonable estimates for U .

$$U = \frac{\rho_v C_{pv} V_v}{(\sum_{m=1}^M A_{jm} + \sum_{n=1}^N A_{cn})(t_f - t_0)} \times \ln \left(\frac{\sum_{m=1}^M A_{jm} [T_v(T_0) - T_{jma}] + \sum_{n=1}^N A_{cn} [T_v(T_0) - T_{cna}]}{\sum_{m=1}^M A_{jm} [T_v(T_f) - T_{jma}] + \sum_{n=1}^N A_{cn} [T_v(T_f) - T_{cna}]} \right) \quad (2.38)$$

Rai et al. (2000) studied for some Newtonian and non-Newtonian fluids mixed in a flat bottomed vessel equipped with a helical ribbon agitator. Heat transfer rates from jacket to bulk liquid were determined for different stirrers, rheological properties and operating conditions. The heat transfer coefficient for the liquid inside the vessel is calculated based on the experimental data and an empirical correlation is developed to predict the heat transfer coefficient for the specified operating conditions. He showed that the heat transfer coefficient is maximum when the impeller is placed little above the centre.

W.A. Khan et al. (2006) they studied fluid flow and heat transfer characteristics of power law fluids across a circular cylinder are investigated using a power law model. This model is based on the fact that both fluids exhibits a region of linear relationship between stress and strain rate when viewed on log log plot. The correlations of heat transfer are developed for both isothermal and isoflux boundary conditions. It is found that pseudoplastic fluids offer less skin friction and higher heat transfer coefficients than dilatants fluids. Further the drag coefficients decrease and heat transfer increases with the decrease in power law index. It was shown that the present results are in good agreement with the available suitable data for the full laminar range of Reynolds number in the absence of free stream turbulence and blockage effects.

Triveni et al. (2008) studied experimentally Heat transfer to Newtonian and non-Newtonian fluids in agitated vessel fitted with impeller and a coil for heating and cooling to determine the heat transfer coefficient of castor oil methyl esters, soap solution, CMC and chalk slurries. He studied effect of impeller geometry, speed and aeration. Generalized Reynolds and Prandtl

numbers are calculated using an apparent viscosity for non-Newtonian fluids. The data is correlated using a Sieder-Tate type equation He found that high values of Nusselt numbers are obtained for non-Newtonian fluids when aeration is coupled with agitation. The contribution of natural convection to heat transfer has been accounted for by incorporating the Grashof number.

Vinogradov et al. (2011) experimentally studied Heat Transfer of Non-Newtonian Dilatant Power Law Fluids in Square and Rectangular Cavities. Steady two-dimensional natural convection in fluid filled cavities is numerically investigated for the case of non-Newtonian shear thickening power law liquids. The conservation equations of mass, momentum and energy under the assumption of a Newtonian Boussinesq fluid have been solved using the finite volume method for Newtonian and non-Newtonian fluids. The computations were performed for a Rayleigh number, based on cavity height, of 105 and a Prandtl number of 100. In all of the numerical experiments, the channel is heated from below and cooled from the top with insulated side-walls and the inclination angle is varied. The simulations have been carried out for aspect ratios of 1 and 4. Comparison between the Newtonian and the non-Newtonian cases is conducted based on the dependence of the average Nusselt number on angle of inclination. It is shown that despite significant variation in heat transfer rate both Newtonian and non-Newtonian fluids exhibit similar behavior with the transition from multi-cell flow structure to a single-cell regime.

A. Debab, N. Chergui1 , K. Bekrentchir and J. Bertrand(2011) The objective of this study is to optimize experimental conditions of agitating a non-Newtonian liquid using experimental design methodology. The measurements of the temperatures have been carried out in a jacketed vessel equipped with Turbine impellers. The rheological properties of aqueous solutions of carboxymethyl cellulose sodium salt had been studied using shear stress/shear rate data. The results of the experimental studies, concerning the effect of the diameter of the impeller, the impeller speed and baffled or unbaffled vessel on the overall heat transfer coefficient have been approximated in the form of equations. Based on the optimization criterion, an agitated vessel equipped with Flat Blade Disc Turbine (FBDT) of diameter ratio $d/D = 0.6$ and baffles is proposed as the most advantageous for heat transfer processes.

In this work, experimental design methodology had been used in order to determine the effect of the impeller diameter, the speed of the impeller and the baffles on the overall heat transfer coefficients in a mechanically stirred tank equipped with a jacket. The film heat transfer coefficient is strongly dependent on the speed of the impeller so that it is proportional to $N^{2/3}$. The rheological characterization of the aqueous solution of CMC employed in the study and

concluded that the concentration of 2% mass represent a non Newtonian behavior in the range of shear rates studied. The fluid behavior index, n , showed no noticeable variation with temperature change but the consistency index, k , decreases with increasing temperature

V T Perarasu, M Arivazhagan, and P Sivashanmugam(2011) Heat transfer studies in a coiled agitated vessel with varying heat input is presented using two agitators namely propeller and disk turbine. Heat transfer rate increases with agitator speed for both the agitators for a given heat input. The heat transfer coefficient also increases with heat input for a given agitator speed for turbine agitator for all the heat inputs, whereas for propeller it is increasing up to a certain value and then decreases. The heat transfer coefficient (heat transfer rate) for turbine agitator is higher than that for propeller for all heat inputs. Empirical correlations separately were formed for each agitator and found to fit the experimental values within range of $\pm 15\%$ for both the agitators.

The following empirical correlations were formulated from experimental data for the propeller and turbine agitator respectively

$$N_u = 0.0534 \left(\frac{\rho N D_a^2}{\mu} \right)^{0.62} \left(\frac{C_p \mu}{k} \right)^{0.27} \left(\frac{D_a}{D_T} \right)^{0.1} \left(\frac{D_o}{D_T} \right)^{0.5} \left(\frac{\mu}{\mu_w} \right)^{0.17} \quad (2.39)$$

$$N_u = 0.0877 \left(\frac{\rho N D_a^2}{\mu} \right)^{0.64} \left(\frac{C_p \mu}{k} \right)^{0.29} \left(\frac{D_a}{D_T} \right)^{0.1} \left(\frac{D_o}{D_T} \right)^{0.5} \left(\frac{\mu}{\mu_w} \right)^{0.21} \quad (2.40)$$

The values obtained from correlation are compared with the experimental values and the deviation is found to be $\pm 15\%$. Figures 4 and 5 present the comparison of correlated and experimental Nusselt number for propeller and disk turbine agitators.

Ashok Reddy K, Bhavanth Rao M, Ram Reddy P (2012) they developed a mathematical model to analyze the heating rates and compare the relation with experimental results obtained in our present study. Low shear rate concentration of sodium carboxymethyl cellulose fluids with two different coil lengths 2.362m and 2.82m, diameter of the helical coil equal to 156mm, $d_i=4.0\text{mm}$ and $d_o=6.4\text{mm}$ were used to correlate overall heat transfer coefficients in an agitated vessel with four blade paddle impeller. The model is derived by using velocity flow field and energy equations in cylindrical coordinate for straight tube and later extended to helical coil. The new design relation for obtaining the individual heat transfer coefficient in terms of flow behavior index is equal to

$$Nu = [(280n^3 + 296n^2 + 88n + 8)/(31n^3 + 43n^2 + 13n + 1)]. \quad (2.41)$$

Pawar et al. (2013) performed experimental studies on Newtonian and non-Newtonian fluids. He used Water, glycerol–water mixture as Newtonian fluids and dilute aqueous polymer solutions of sodium carboxymethyl cellulose (SCMC), sodium alginate (SA) as non-Newtonian fluids for his study. He tried to correlate Nusselt number to dimensionless number for Newtonian fluids based on experimental data. The comparison of overall heat transfer coefficient U_o , and Nusselt numbers for Newtonian and non-Newtonian fluids under isothermal and non-isothermal conditions is studied. From the experimental results, it was found that overall heat transfer coefficient and Nusselt numbers for water are higher than glycerol–water mixture and non-Newtonian fluids. It was also observed from experimental results that as helix diameter increases, overall heat transfer coefficient and Nusselt numbers of both fluids decreases for the same flow rates.

In this study, they simulated and analyzed the convective heat transfer of a non-Newtonian nanofluid in the turbulent flow of a circular tube using fluent software. The effect of particles and Reynolds number on local, convective heat transfer and Nusselt number was investigated.

S.S. Pawar, Vivek K. Sunnapwar (2013) experimentally studied on isothermal steady state and non-isothermal unsteady state conditions were carried out in helical coils for Newtonian and non-Newtonian fluids. Water, glycerol–water mixture as Newtonian fluids and dilute aqueous polymer solutions of sodium carboxymethyl cellulose (SCMC), sodium alginate (SA) as non-Newtonian fluids were used in this study. These experiments were performed for coils with curvature ratios $d = 0.0757, 0.064$ and 0.055 in laminar and turbulent regimes. An innovative approach of correlating Nusselt number to dimensionless number ‘M’ for Newtonian fluids based on experimental data is presented which is not available in the literature. Several correlations for the first time are proposed based on heat transfer data generated from the experiments performed for Newtonian fluids under isothermal and non-isothermal conditions (total 138 tests). These developed correlations were compared with the work of earlier investigators and were found in good agreement. Further, comparison of overall heat transfer coefficient U_o , and Nusselt numbers for Newtonian and non-Newtonian fluids under isothermal and non-isothermal conditions (total 276 tests) is presented in this paper. From the experimental results, it was found that overall heat transfer coefficient and Nusselt numbers for water are higher than glycerol–water mixture and non-Newtonian fluids. It was also observed from experimental results that as helix diameter increases, overall heat transfer coefficient and Nusselt numbers of both fluids decreases for the same flow rates.

The results in this work showed that overall heat transfer coefficient is higher for smaller helix diameter as compared to larger helix diameter due to significant effect of centrifugal force on secondary flow in coil. It is also observed from results that heat transfer coefficients for pure water are higher than glycerol–water mixture and non-Newtonian fluids for the same conditions. In future work, Nusselt number need to be correlated to M number based on experimental data for any type of power-law fluids and any coil curvature ratios, and shall be validated with the work of earlier investigators.

Mohammad Sharifi Asl.et.all.(2014) attempted to simulate the turbulent flow of a non-Newtonian nanofluid in a circular, horizontal tube with regard to constant heat flux on the tube wall, using Computational Fluid Dynamics (CFD). To this effect, using fluent software, the conservation equations of momentum and energy for turbulent flow of a non-Newtonian fluid and a nanofluid containing alumina particles in the nonNewtonian fluid is analyzed. An aqueous solution of carboxymethyl cellulose (CMC) was used as the base non-Newtonian fluid. Temperature field of nanofluids was obtained and, analyzing the results, the heat transfer coefficient of nanofluids was calculated. The results obtained were compared with the results of the base fluid. The effects of volume fraction or nanoparticle concentration and Reynolds number on heat transfer coefficient and Nusselt number were also investigated. The results indicated an increase in heat transfer coefficient and Nusselt number using non-Newtonian nanofluid compared to the base non-Newtonian fluid. This increase had a direct relationship with the volume fraction of the nanoparticles and the Reynolds number. The results showed good agreement with the results of laboratory research.

N.Fedal Castro1, B.Chitra, R.Pushpalatha, S.Sudalai (2014) In many chemical process industries, mixing and heat transfer in agitated vessel is an important operation in both batch and continuous process. Agitated vessels are generally used for processing liquid systems. In this project work, the effect of mixing in heat transfer from jacket to bulk of liquid in an agitated vessel have been investigated both for Newtonian (water) and non-Newtonian fluid (Carboxy methyl cellulose solution 0.5 and 1.5 weight % concentration). The experiments were conducted in an unbaffled dished bottom vessel using two different impellers separately, one is a four bladed paddle type impeller having a diameter of 0.314 m and the other type is V-shaped impeller of diameter 0.0775 m. An empirical correlation is developed to calculate heat transfer coefficient for both Newtonian and Non-Newtonian fluid using two different impellers. Investigation also includes the comparison of power number for the impellers.

Power number was also calculated for Newtonian and Non-Newtonian fluids using two different impellers. A generalized correlation was found for power number by plotting a graph between \ln (Power number) and \ln (Reynolds number). We obtained the following results. This experiment confirms that the impeller was effective in mixing and with increase in mixing effect heat transfer coefficient enhances. From the experimental work using paddle type impeller and V shaped impeller we conclude that V type impeller was effective in mixing both Newtonian and Non-Newtonian fluids with least power consumption

CHAPTER 3

MATERIALS AND METHODOLOGY

3.1 Heat Transfer from Vessel Wall to the Agitated Pseudo plastic Fluids

The earlier approaches to correlate heat transfer data on power law type non-Newtonian fluids in agitated vessels may be categorized as follows:

Carreau, Charest and Corneille (1966) studied heat transfer to pseudo plastic fluids in jacketed vessels using turbine agitators and employed a generalized Reynolds number analogous to that of laminar flow in pipes. The flow behavior of the fluid in the agitated vessel is not adequately described by the Reynolds number used by the above workers. Prandtl number has been defined by using a differential viscosity at high shear rate, i.e.

$$N_{Pr_{d\infty}} = \frac{C_p \mu_{d\infty}}{k} \quad 3.1.1$$

Where

$$\mu_{d\infty} = \left[\frac{d\tau}{d} \left(\frac{du}{dr} \right) \right]_{\infty} \quad 3.1.2$$

And Reynolds number uses pipe flow viscosity

$$N'_{Re} = \frac{D_a^2 N^{2-n} \rho}{\dot{\tau}} \quad 3.1.3$$

Where

$$\dot{\tau} = \frac{g_c K}{8} \left(\frac{6n+2}{n} \right)^n \quad 3.1.4$$

Thus in the final correlation some variation exists in the coefficients for Newtonian and non-Newtonian fluids. The second approach has been presented by Gluz and Pavlushenko (1966) in which they have used the viscosity

$$\mu_{av} = K(\dot{\tau}_{av})^{n-1} \quad 3.1.5$$

Where

$$\dot{\tau}_{av} = 4\pi N \quad 3.1.6$$

In the Reynolds and Prandtl numbers. The average shear rate ($\dot{\gamma}_{av}$) corresponds to the shear rate at the surface of the rotating cylinder, and the Reynolds number defined by these workers describes the flow behavior in a better manner compared to that used by Carreau *et al.* (1966). Hagedorn and Salamone (1967) considered the flow pattern at the vessel wall for power law fluids and presented the correlation on the basis of dimensional analysis of basic equations. Mizushima, Ito, Murakami and Tanaka (1966) also considered the shear rate at the wall and correlated the data on Newtonian and non-Newtonian fluids for anchor agitator assuming the shear rate proportional to the ratio of agitator tip speed and the clearance between the agitator tip and the wall. Many other workers have made efforts to correlate the similar data by using average shear rate derived from agitator power measurements following the method of Metzner and Otto (1957) and Calderbank and Moo Young (1957, 1959).

Ashok Reddy K , Bhavanth Rao M, Ram Reddy P(2012) they developed a mathematical model to analyze the heating rates and compare the relation with experimental results obtained in our present study.

Pawar et. al. (2013) performed experimental studies on Newtonian and non-Newtonian fluids. He used Water, glycerol–water mixture as Newtonian fluids and dilute aqueous polymer solutions of sodium carboxymethyl cellulose (SCMC), sodium alginate (SA) as non-Newtonian fluids for his study. He tried to correlate Nusselt number to dimensionless number for Newtonian fluids based on experimental data. The comparison of overall heat transfer coefficient U_o , and Nusselt numbers for Newtonian and non-Newtonian fluids under isothermal and non-isothermal conditions is studied.

3.2 Present Approach

For power law fluids, shear stress and shear rate relationship at the rotor of the bob viscometer in an infinite fluid (considered as the surface of the cylinder rotating in the infinite fluid) is given by

$$\tau_s = \left(r \frac{d(u_\theta/r)}{dr} \right)_s^n = K \left(\frac{4\pi N}{n} \right)^n \quad 3.2.1$$

And the relationship for the generalized power law may be stated as

$$\tau_s = K'' (4\pi N)^{n''} \quad 3.2.2$$

$$\left[r \frac{d(u_o/r)}{dr} \right]_s = \frac{4\pi N}{n''} \quad 3.2.3$$

$$\text{And } K'' = \frac{K}{(n)''} \quad 3.2.4$$

For power law fluids ($n'' = n$) various viscosity and Reynolds number expressions may be defined as follows:

Pseudo shear viscosity

$$\mu_e'' = \frac{\tau_s}{4\pi N} = \frac{K}{(n)''} (4\pi N)^{n-1} \quad 3.2.5$$

And the corresponding Reynolds number

$$N_{Re}'' = \frac{D_a u_s \rho}{\mu_e} = \frac{(n)'' D_a^2 N^{2-n} \rho}{K(4)''^{n-1} (\pi)''^{n-2}} \quad 3.2.6$$

Apparent or shear viscosity is written as

$$\mu_a'' = \frac{\tau_s}{\left[r \frac{d(u_\theta/r)}{dr} \right]_s} = n^{1-n} K (4\pi N)^{n-1} \quad 3.2.7$$

And

$$N_{Rea}'' = \frac{D_a^2 N^{2-n} \rho}{n^{1-n} K(4)''^{n-1} (\pi)''^{n-2}} \quad 3.2.8$$

Differential viscosity

$$\mu_d'' = \frac{d\tau_s}{d \left[r \frac{d(u_\theta/r)}{dr} \right]_s} \quad 3.2.9$$

Equation (3.2.9) may be written as

$$\mu_d'' = \frac{\tau_s}{r \left[r \frac{d(u_\theta/r)}{dr} \right]_s} \cdot \frac{d \ln \tau_s}{d \ln \left[r \frac{d(u_\theta/r)}{dr} \right]_s} n \mu_a'' \quad 3.2.10$$

From equation (3.2.8), (3.2.9) and (3.2.10) one obtains

$$\mu_d'' = n \mu_a'' n^2 \mu_e'' \quad 3.2.11$$

The relationship between the various Reynolds numbers N''_{Re} , N''_{Rea} and N''_{Red} on the basis of pseudo shear, shear and differential viscosities, respectively may be written as

$$N''_{Re} = nN''_{Rea} = n^2 N''_{Red} \quad 3. 2.12$$

Similarly corresponding Prandtl numbers may also be defined as

$$N''_{Prd} = nN''_{Pra} = n^2 N''_{Pr} \quad 3. 2.13$$

Where

$$N''_{Pr} = \frac{C_p K (4\pi N)^{n-1}}{(n)^n k_t} \quad 3. 2.14$$

The analyses of the flow pattern around the mixing impellers are not so simple. A compromise between experimental and physical pictures of the flow pattern is needed to obtain suitable parameters for the correlation. Let us consider the case of a flat plate submerged in a fluid flowing in a laminar fashion. If the thickness of the flat plate is small compared to width, most of the drag on the plate will be due to the pressure difference between the front and the rear of the plate which is caused by the motion of the fluid near the surface. Shear stress causing any drag will be negligible. Now let us consider the flow situations around the flat blade of a turbine or paddle rotating in a fluid. At low speed of rotation the flow pattern may be considered similar to that of a flat plate submerged in a fluid, in motion. At higher speed, with more number of blades, the extent of the separation of flow will decrease and the viscosity energy dissipation becomes the controlling factor. Thus the use of dimensionless groups defined by visualizing the system as a cylinder of diameter equal to that of blade rotating in an infinite fluid medium will be better as compared to the paradoxical definitions of Reynolds number and viscosity analogous to pipe flow.

Now, let us consider the heat transfer from the wall of the vessel agitated by a turbine agitator. Fluid flow along the wall of the vessel is upward and resistance to heat transfer is mainly controlled by the viscous sub layer along the wall, with a large velocity and temperature gradient in radial direction. The heat transferred through the viscous sub layer is completely mixed with the bulk of the fluid between the sub layer and the impeller. Under steady state conditions the momentum, mass and energy equations for power law fluids in the region near the wall may be simplified and presented in the following form:

$$\rho u_r \left(\frac{\partial u_z}{\partial r} \right) = \frac{\partial p}{\partial z} - \frac{1}{r} \cdot \frac{\partial}{\partial r} \left[-rK \left(\frac{\partial u_z}{\partial r} \right)^n \right] + \rho g_z \quad 3.2.15$$

$$\frac{1}{r} \cdot \frac{\partial}{\partial r} (\rho r u_r) = 0 \quad 3.2.16$$

$$u_r \frac{\partial T_i}{\partial r} = \frac{k_t}{\rho c_p} \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T_i}{\partial r} \right) \right] \quad 3.2.17$$

The viscous sub layer thickness δ will be very small compared to the vessel diameter. As Reynolds number, N''_{Rea} , increases the thickness of the viscous sublayer decreases. Thus δ will be a function of D_a and N''_{Rea} and $u_{r\delta}$ will be a function of N''_{Rea} and characteristic velocity, $\pi D_a N$.

The following dimensionless variables, now, can be defined as

$$r^* = \left(r/D_a \right) N''_{Rea}{}^m \quad 3.2.18$$

$$u_r^* = \left(u_r/\pi D_a N \right) N''_{Rea}{}^m \quad 3.2.19$$

$$\text{As } z^* = z/D_a, \quad u_z^* = u_z/\pi D_a N \quad 3.2.20$$

$$P^* = \frac{(P - P^0)}{\rho(\pi D_a N)^2} \quad 3.2.21$$

$$T^* = \frac{(T_i - T)}{(T_s - T)} \quad 3.2.22$$

Substituting these dimensionless variables in equations (3.2.21) and (3.2.22), it is observed that

the velocity gradient $\left(\frac{\partial z^*}{\partial r^*} \right)$ is a function of $(N''_{Rea})^{m(n+1)-1}$ and the temperature gradient

$\frac{\partial T^*}{\partial r^*}$ is a function of $(N''_{Rea})^{2m-1} / N''_{Pra}$.

Using Fourier's law of heat conduction and Newton's law of cooling, the expression for the total heat flux and heat transfer coefficients can be obtained. The final expression for Nusselt number may be arranged as

$$N_{Nu_j} = \frac{h_j D_T}{k_t} = \frac{D_T N''_{Rea}{}^m}{H} \int_0^{H/D_a} \left(\frac{\partial T^*}{\partial r^*} \right)_w dz^* \quad 3.2.23$$

Thus it is seen that N_{Nuj} is a function of N''_{Rea} , N''_{Pra} and length ratio (H/D_a) . If non-isothermal correction factor is included, the correlation takes the form,

$$N_{Nuj} = C_1 N''_{Rea}{}^{b_1} N''_{Pra}{}^{b_2} (\mu_{ab}/\mu_{aw})^{b_3} \quad 3.2.24$$

Retaining the conventional values of Reynolds and Prandtl number indices as $b_1 = 2/3$ and $b_2 = 1/3$, respectively and Sieder-Tate correction factor $(\mu_{ab}/\mu_{aw})^{0.14}$, the equation (3.2.24) takes the following shape

$$N_{Nuj} = C_1 (N''_{Rea})^{2/3} (N''_{Pra})^{1/3} (\mu_{ab}/\mu_{aw})^{0.14} \quad 3.2.25$$

For isothermal heat transfer (μ_{ab}/μ_{aw}) will be almost equal to 1.

3.3 Heat Transfer from Agitated non-Newtonian Fluids to Helical Coils

Several investigators have studied the average heat transfer from helical coils in the mixing vessels. The heat transfer rate at the coil surface may be written as

$$Q = -k \int_A \left(\frac{\partial T_i}{\partial x} \right)_s dA \quad 3.3.1$$

Where A is the total surface area and x is the distance measured outward from and normal to the coil surface. Flow from the impeller passing around the coil surface, forms boundary layer which in turn offers resistance to heat transfer. Thus the velocity and temperature profile will depend on the impeller Reynolds number, Prandtl number and the diameter of the coil tube. On the basis of the discussions made in the previous section it can be shown that the temperature gradient in equation (3.3.1) is a function of (N''_{Rea}) and (N''_{Pra}) . Thus, in order to describe the heat transfer from the coil surface, a correlation can be developed between the Nusselt number $\left(h_{oc} \frac{D_t}{k} \right)$, Reynolds number, (N''_{Rea}) , and Prandtl number, (N''_{Pra}) . For coils in mixing vessel and for given geometrical situation and isothermal condition this results in the following equation

$$\frac{h_{oc} D_o}{k} = C_2 (N''_{Rea})^{b_3} (N''_{Pra})^{b_4} \quad 3.3.2$$

3.4 Laminar Forced Convection to non-Newtonian Fluids in coiled Tube

Laminar forced convection in curved pipes has received considerable attention in recent years. From the work of Akiyama and his co-workers (1971), Dravid (1971), Kubair and Kuloor (1966) and Rajshekhran et al. (1966), it is seen that very little work has been done for non-Newtonian fluids in curved pipes. The existing heat transfer results in the literature for fully developed laminar forced convection in curved pipe with uniform wall temperature are rather limited and incomplete. For Newtonian fluids, perturbation method was applied by Maekawa for extremely low Dean Number. The boundary layer approximation near the wall was presented by Mori and Nakayama (1965) for high Dean Number of order one. Dravid et al. (1971) presented the numerical results in the thermal entrance region for Dean Number less than 225 and $N_{pr} = 5$. The recent and improved results of Akiyama (1971) show that the ratio of heat transfer coefficients in coil and in straight pipe is a function of Dean Number as well as Prandtl number.

Thus there is possibility of obtaining a suitable correlation of the following form

$$\frac{N_{Nuic}}{N_{Nuis}} - 1 = \phi_1 (N_D, N_{Pr}) \quad 3.4.1$$

The advantage of the above form of correlation is that for small N_D , the equation (3.4.1) satisfies the condition that $N_{Nuic} \approx N_{Nuis}$ for very very small N_D . N_{Nuis} may be calculated for isothermal laminar heat transfer in straight tube for the case of uniform wall temperature and parabolic velocity distribution by the following correlation:

$$N_{Nuis} = 1.75 (N_{GZ})^{\frac{1}{3}} \quad 3.4.2$$

For non-Newtonian fluids equations (3.4.1) and (3.4.2) take the following form:

$$N_{Nuis} = 1.75 \left(\frac{3n+1}{4n} \right) (N_{GZ})^{\frac{1}{3}} \quad 3.4.3$$

Here again it is important to note that Dean number contains Reynolds number and the viscosity term appears in both the Reynolds and Prandtl numbers. It is suggested that the effective viscosity μ_2 at the shear stress prevailing at the wall as discussed in chapter 2 should be used for evaluating the Reynolds and Prandtl numbers. From the above discussions it may be concluded that the correlation may be written as below:

$$\frac{N_{Nuic}}{N_{Nuis}} - 1 = C_3 N_{D_2}^{b_5} N_{Pr_2}^{b_6} \quad 3.4.4$$

3.5 MEASUREMENT OF RHEOLOGICAL PROPERTIES

A capillary tube viscometer, as shown schematically in figure 3.5.1 was designed to meet the following requirements and limitations imposed by the systems studied:

1. Evaluating of shearing stresses and rates of shear to determine the pseudo shear curves for various fluids.
2. Operation at pressure drops up to 4.0 kgf/cm² and suitable to provide the entire range of shear stresses wherein measurements were made.
3. Operation within a temperature range of 20 to 80⁰C within ± 0.1 ⁰C limit.
4. Be relatively easy to dismantle for cleaning etc.

3.5.1 Viscometer

The capillary tube viscometer consisted of a cylindrical, brass reservoir of 2 liters capacity at the base of which a Pyrex-glass precision bore capillary tube was fastened by a hand tightened cap. The diameter and the length of the capillary tubing were carefully selected such that the maximum variation did not exceed 0.2% in 1 mm diameter size and that the length to diameter ratio was more than 500.

Since temperature has appreciable effect in a constant temperature bath. The bath consisted of a rectangular vessel made from 1 inch thick Perspex sheet and was completely filled with water so that the brass reservoir was completely submerged into it. The capillary tube protruding outside the reservoir was surrounding by $\frac{3}{4}$ inch diameter Perspex tubing which in turn was connected to the bottom of the rectangular vessel. Circulation of water from the bath through the Perspex tubing facilitated to maintain a constant and uniform temperature of the test fluid both in the reservoir and the capillary. Agitators were used to agitate the fluid in the reservoir and the bath to further ensure temperature uniformity throughout the system.

A reciprocating compressor supplied air to a tank of 5 cu ft capacity which was transferred to the reservoir through a three way valve which also vented the reservoir. The pressure was controlled by a pressure regulator and measured by a calibrated gauge or a mercury manometer of 200 cm length.

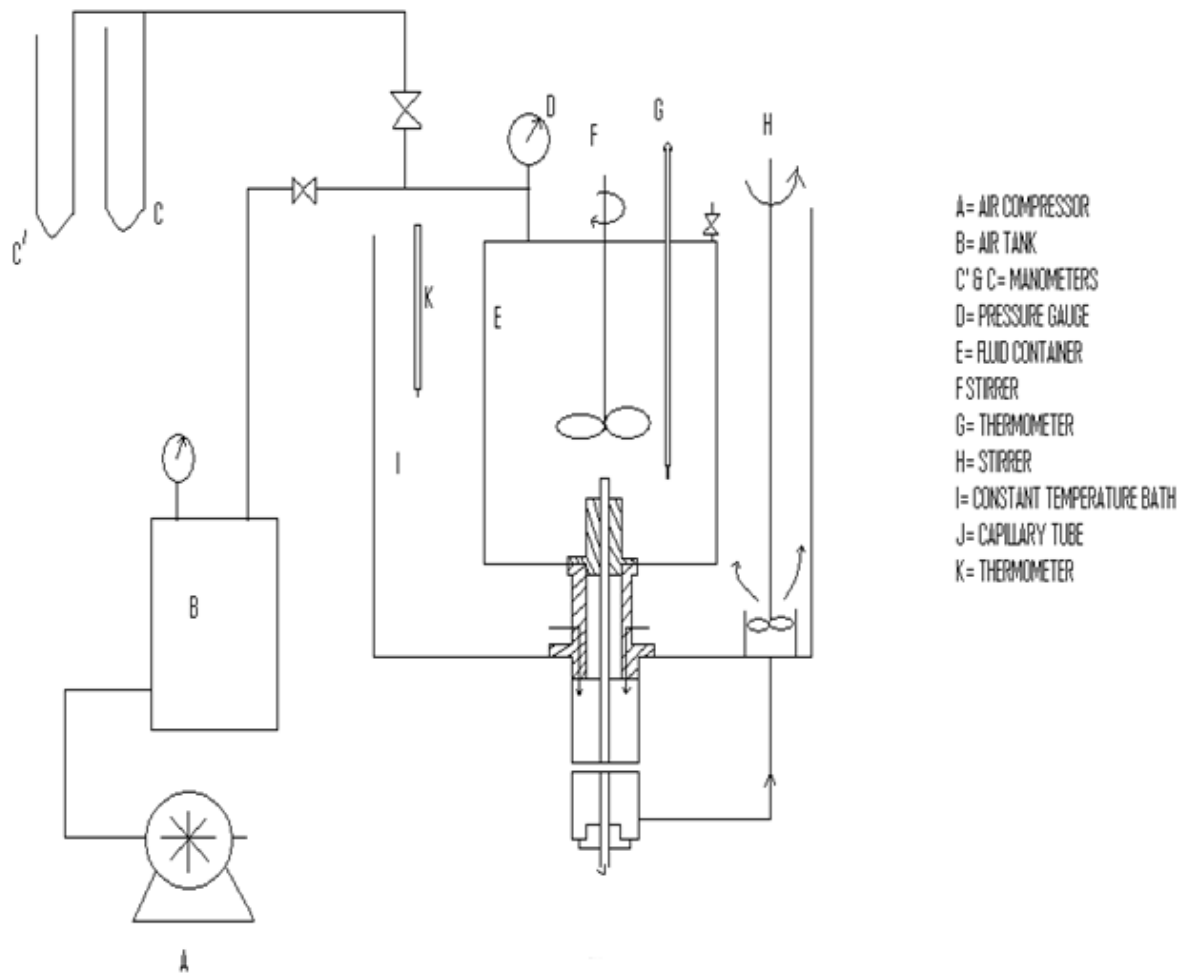


FIG 3.5.1 : Capillary Tube Viscometer

Two electric heaters each of 100 watts, connected with a temperature controlled, were used to maintain the desired temperature and these temperatures were recorded by two thermometers, both capable of reading up to $1/10^{\text{th}}$ of a degree centigrade.

3.5.2 Procedure

Data were obtained by closing the end of capillary tube with a stopper and filling the reservoir to a known height with the Newtonian or non-Newtonian fluid in question. The temperature in the water bath and the fluid reservoir was maintained in a desired value with the help of heaters and stirrers. The pressure cap on the top of the reservoir was then screwed into the place; the pressure was adjusted in the pressure vessel to the desired value, and then applied to the reservoir via the three way valve. The stopper was removed from the capillary tube and about 25 to 50 cc of fluid was collected in the collecting tank during a known interval of time. This material was weighted and the corresponding volumetric flow rate was calculated from knowledge of the density of the fluid. The applied differential pressure was varied from determination to determination so that the time of flow varied from approximately 20 seconds to 1000 seconds, or so that the smallest differential pressure was approximately 4 cm Hg. The diameter of the capillary tube was measured by mercury filling method.

The actual measurements taken are: temperature, gas pressure in the reservoir, height of the fluid in the reservoir, mass or volume of the fluid flowing through the capillary tube and the corresponding time interval of collection, capillary tube diameter and length.

Five to eight readings at different flow rates or at different differential pressure were taken for each temperature. These sets of observations at two to four temperatures were taken for each fluid.

3.5.3 Calibration

The capillary tube viscometer was standardized with a standard viscosity fluid. An oil Brookfield viscosity standard fluid-500 of viscosity 504 centipoises at 25⁰C, Lot No. 120270, was chosen as standard fluids of which viscosity-temperature data were available. Physical dimensions of the capillary tube were verified to $\pm 0.5\%$ accuracy.

The fluids for which rheological data were obtained are glycerin-water solution, six aqueous carboxy methyl cellulose solutions and one aqueous polyvinyl alcohol solution. The samples of these solutions were obtained directly from the fluid prepared for pressure drop or heat transfer studies.

The fluids studied and their physical properties are listed in table 3.6. Viscosities and other flow constants of the fluids were determined as described below.

3.5.4 Determination of the Pseudo-shear Curve or the Plot $D \Delta P/4L$ versus $8U/D_t$

This plot is the first step in finding out the viscosities or flow constants of the purely viscous non-Newtonian fluids and it evidently follows directly from the measurements in the case of a capillary tube viscometer. The total mechanical energy balance equation (including pressure energy, potential energy, kinetic energy and correlations for entrance effects) for the capillary tube viscometer is as follows:

$$\Delta P = P_{gas} - P_{atm} + (L + L')\rho \frac{g}{g_c} - 1.12 \frac{\rho U^2}{g_c} \quad 3.5.4.1$$

Where L = length of the capillary tube

L' = fluid level above the top of the capillary tube in the reservoir and

$1.12 \frac{\rho U^2}{g_c}$ is the correction due to the kinetic energy and entrance effects.

The uncertainties regarding the correction term are minimized by making L/D_t ratio for the capillary tube as large as 600 to 700. This correction was found to be negligible for the majority of the viscometric data taken in the present study.

From the diameter D_t and length L , fluid level L' , gas pressure, P_{gas} , and flow rate from the capillary, the shear stress at the wall or τ_w ($D_t P/4L$) were calculated. Pseudo-shear rate ($8U/D_t$) were also calculated from the velocity and diameter of the tube. The measurements made on the capillary tube viscometer were thus converted to a logarithmic plot of τ_w versus $8U/D_t$. The slope of the curve at a particular shear stress at the wall, n' , were evaluated according to the generalized power law equation given below:

$$\tau_w = K' \left(8U/D_t\right)^{n'} \quad 3.5.4.2$$

once the value of n' was known the corresponding value of K' was calculated by the above equation and thus a plot of K versus temperature could be obtained after further calculations.

All the CMC solutions were considered as pseudoplastic fluids obeying power law relation

$$\tau = K \left(-\frac{du}{dr}\right)^n$$

Behavior of these solutions and their shear diagrams are shown in figure 3.5.4.A through 3.5.4.D. The flow behavior index, 'n', at various temperatures was found to be the same for a particular fluid whereas the consistency index, K, decreased with the temperature as shown in figure 3.5.4.E. Three aqueous solutions containing 0.5 (n= 0.937), 1 (n= 0.851) and 2% (n=0.793) CMC-A and water were studied in the coil and 1,2 and 4% (n=0.698) aqueous solutions of CMC-A in the vessel. The thermal conductivities of CMC-A solutions were determined by comparative concentric cylinders method described in chapters and were found to be very near or equal to that of water. The specific heats of these fluids were also found to be equal to that of water and no appreciable change was seen due to varying polymeric concentration in the water.

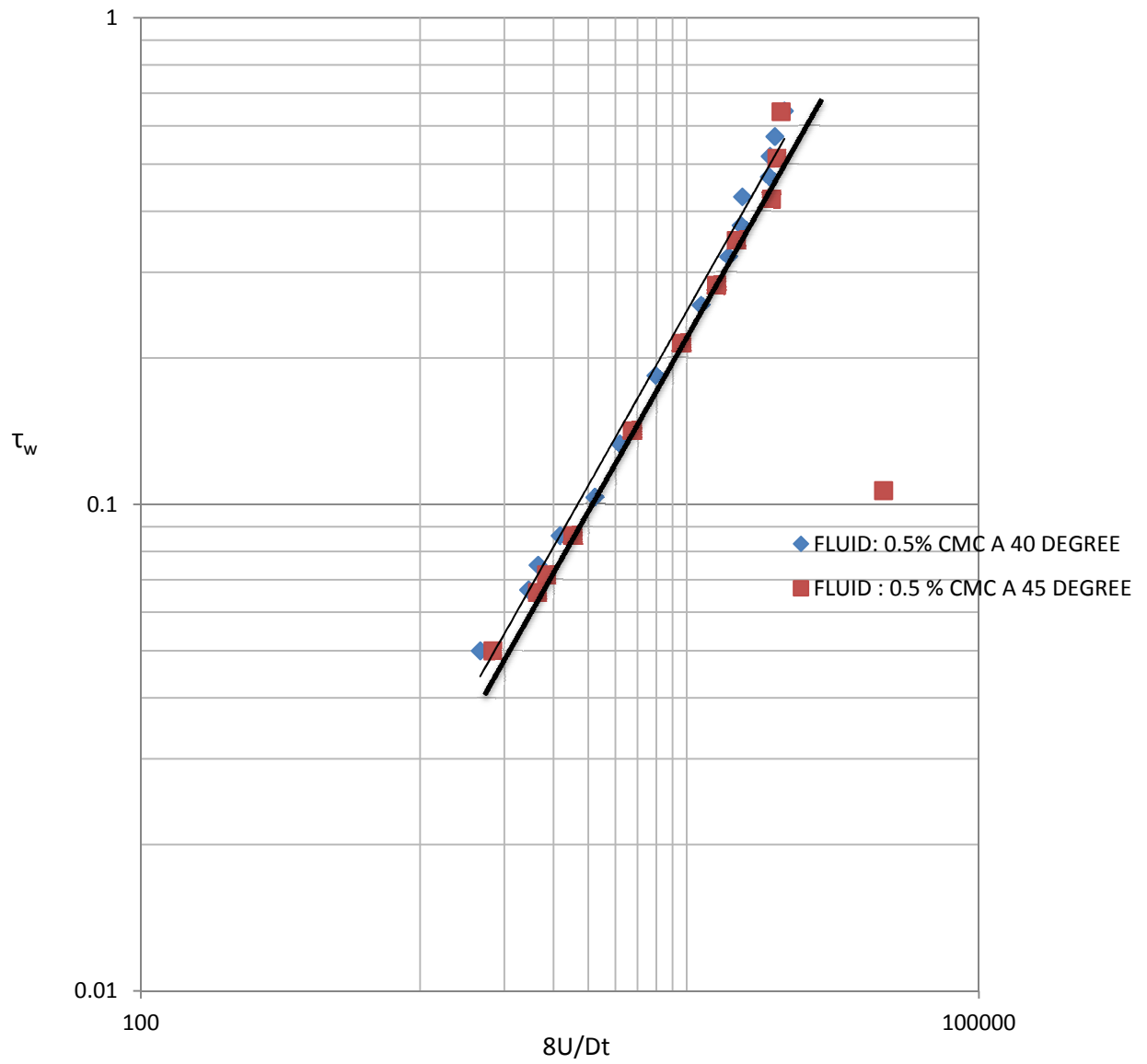
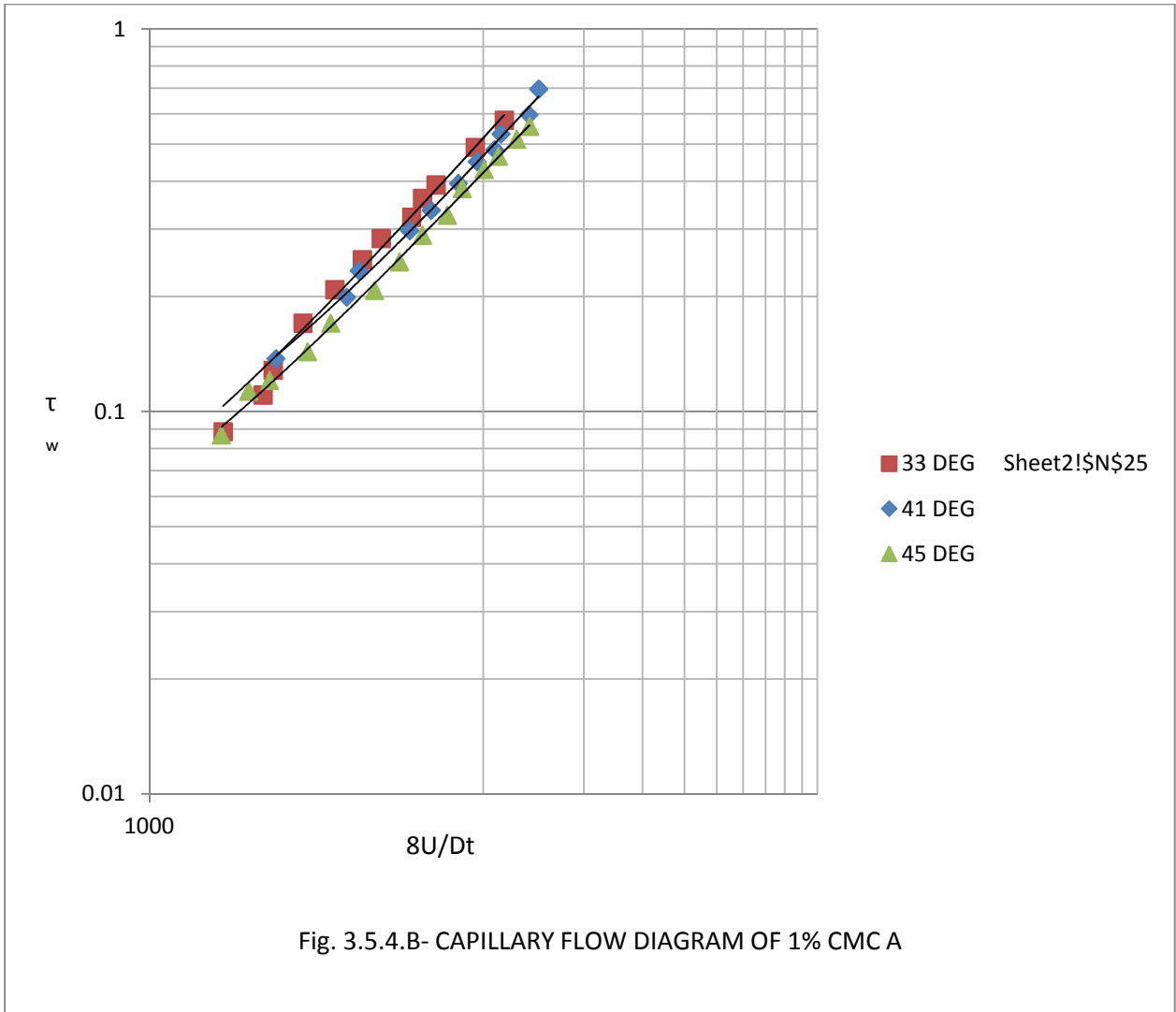


Fig. 3.5.4.A - CAPILLARY FLOW DIAGRAM OF 0.5% CMC A



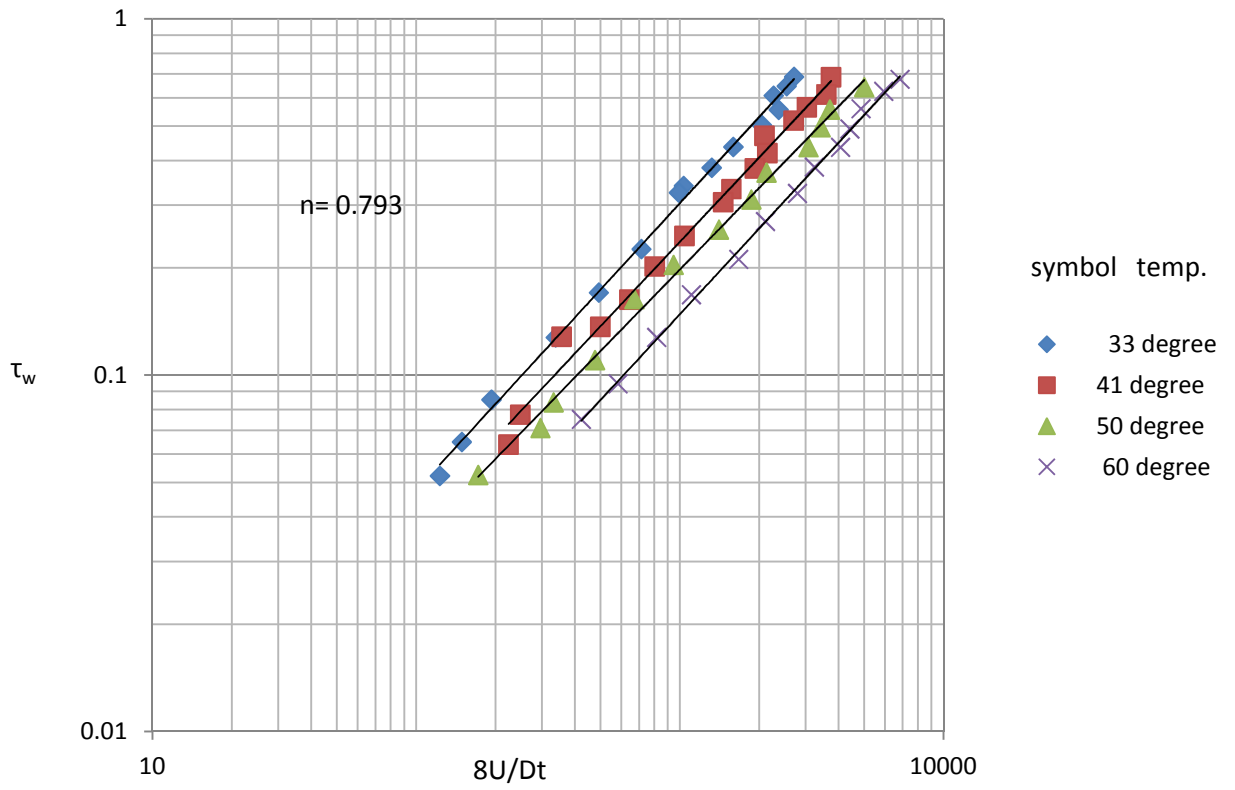


Fig. 3.5.4.C- CAPILLARY FLOW DIAGRAM OF 2% CMC-A

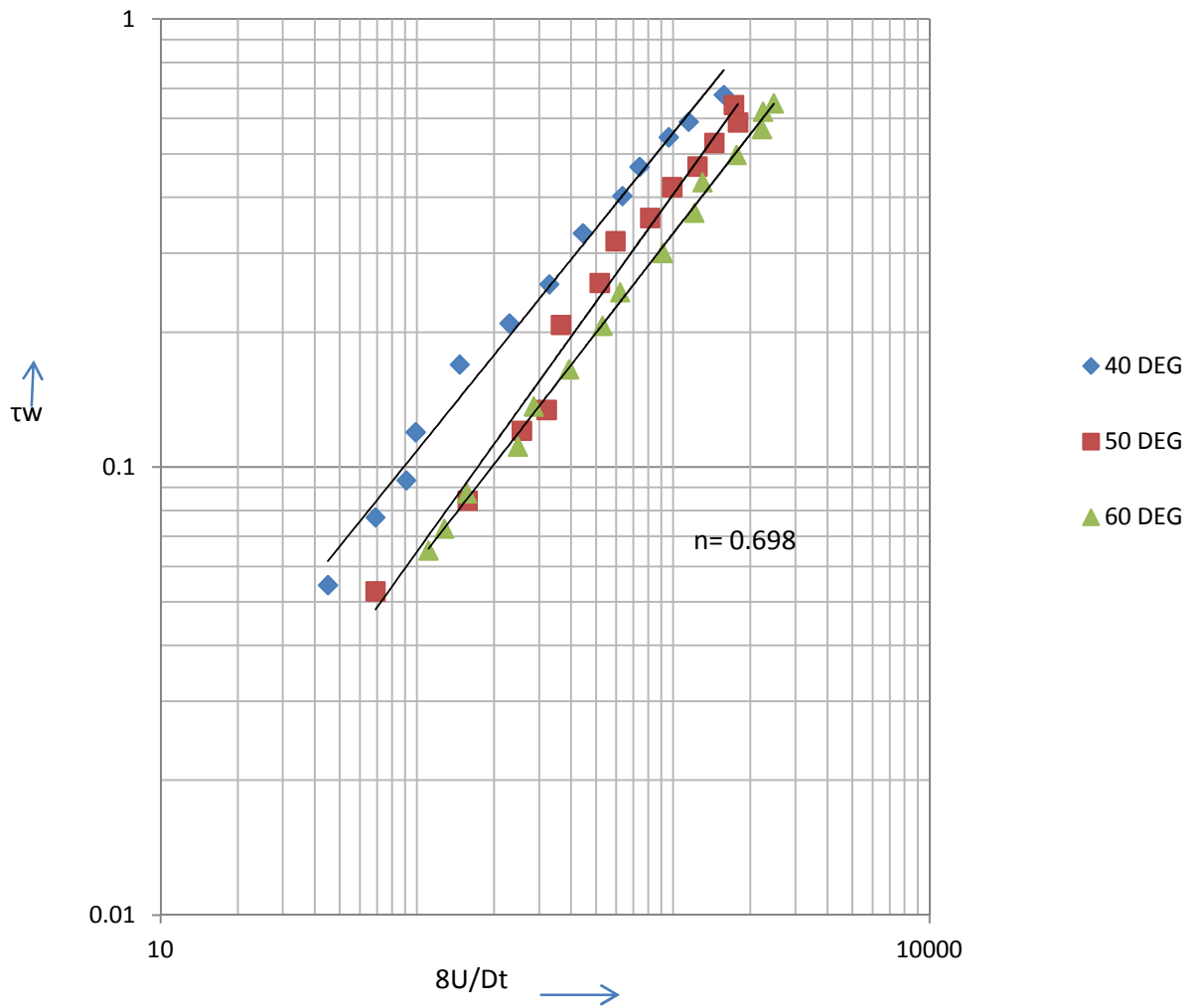


FIG.3.5.4.D- CAPILLARY FLOW DIAGRAM OF 4% CMC A

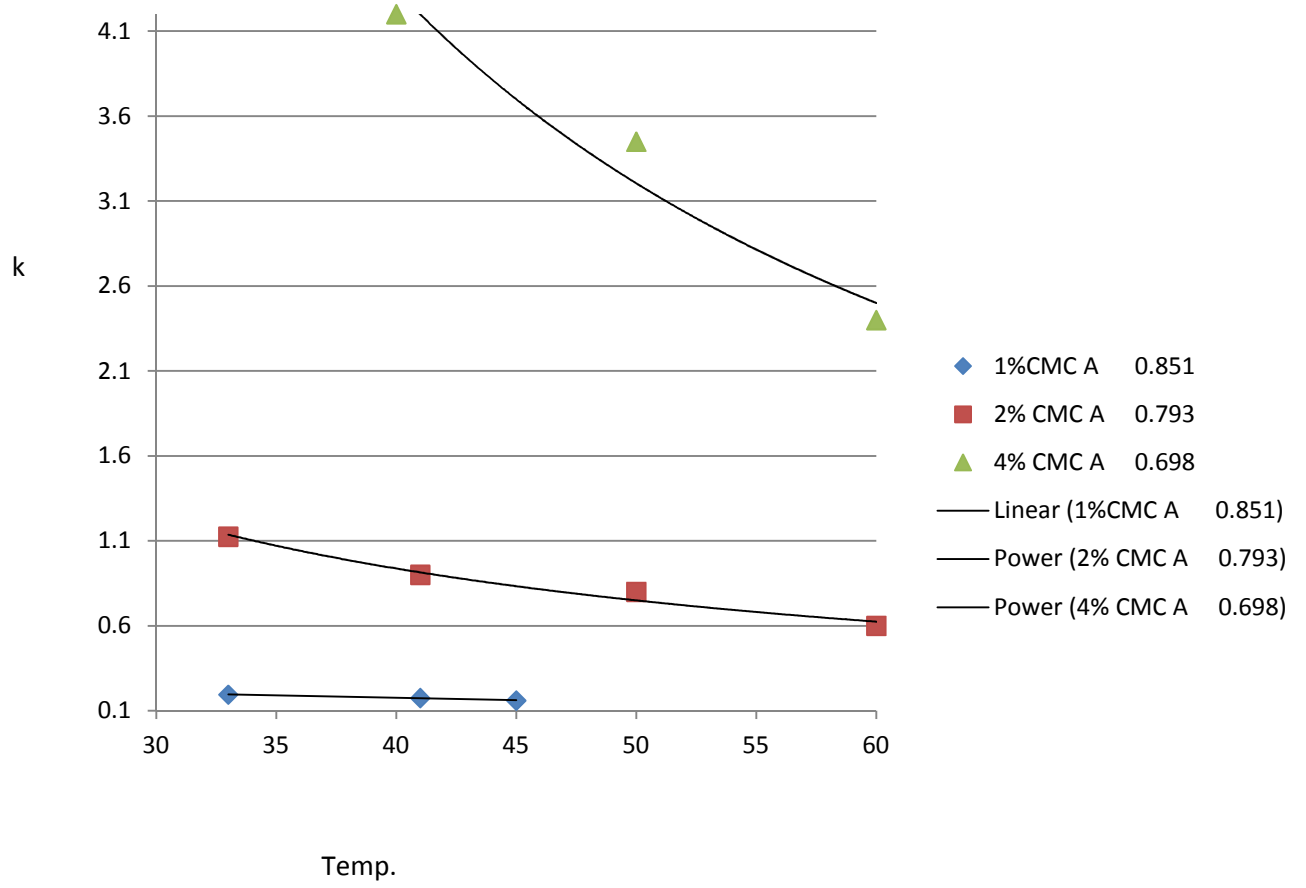


Fig. 3.5.4.E- VARIATION OF FLOW CONSISTENCY WITH TEMPERATURE

TABLE 3.6.1 RHEOLOGICAL PRPPERTIES

Capillary Tube I.D.= 0.17 cm , L = 67.35 cm

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 0.5% CMC A		Temperature = 40°C		
1	1038.78	0.64404	427.91	20137
2	920.012	0.5704	395.81	18626
3	837.96	0.51953	380.026	17884
4	760.21	0.47133	377.91	17784
5	691.758	0.42888	302.733	14246
6	604.266	0.37464	299.47	14093
7	520.854	0.32292	270.238	12717
8	414.322	0.25678	215.079	10129
9	296.91	0.18408	148.236	6976
10	214.854	0.13321	110.229	5187
11	166.806	0.10341	89.636	4218
12	139.154	0.08627	67.358	3170
13	121.022	0.07503	56.186	2644
14	107.65	0.06674	51.97	2446
15	80.678	0.05002	34.88	1641
Fluid= 0.5% CMC A		Temperature = 45°C		
1	80.678	0.05002	38.7	1821
2	106.29	0.06589	55.82	2627
3	115.582	0.07166	60.185	2832
4	139.154	0.08627	74.779	3519
5	172.246	0.10679	96.9	45599
6	228.458	0.14164	122.089	5745
7	345.87	0.21443	183.29	8625
8	455.12	0.28217	244.7	11515
9	561.65	0.34882	287.9	13548
10	683.826	0.42397	384.7	18104
11	830.478	0.51489	402.55	18943
12	1034.93	0.64165	416.225	19587

TABLE 3.6.2 RHEOLOGICAL PRPPERTIES

Capillary Tube I.D.= 0.17 cm , L = 67.35 cm

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 1% CMC A		Temperature = 33 ⁰ C		
1	931.70	0.5776	245.85	11569
2	791.6	0.4908	201.135	9465
3	631.2	0.3913	153.500	7223
4	582.2	0.3609	139.800	6579
5	519.6	0.3221	129.700	6104
6	457.1	0.2834	105.240	4952
7	420.6	0.2496	92.400	4348
8	336.07	0.2083	76.420	3596
9	274.8	0.17037	61.310	2885
10	206.86	0.12825	49.950	2350
11	178.30	0.1105	46.550	2191
12	142.95	0.0886	35.37	1664
Fluid= 1% CMC A		Temperature = 41 ⁰ C		
1	1059.59	0.6969	311.5	14659
2	963.03	0.59707	291.4	13713
3	858.3	0.53210	240.1	11298
4	780.70	0.48400	229.95	10821
5	726.3	0.45030	203.75	9588
6	636.8	0.39480	179.0	8423
7	541.8	0.33590	148.4	6983
8	470.6	0.29710	128.07	6027
9	376.8	0.23360	90.24	4246
10	321.1	0.19900	82.90	3901
11	221.8	0.1375	50.90	2395

Contd.....Table 3.6.2

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 1% CMC A		Temperature = 45°C		
1	899.113	0.55740	292.57	13768
2	831.113	0.51529	267.68	12596
3	750.873	0.46550	235.807	11097
4	693.753	0.43012	213.900	10066
5	617.593	0.38290	181.920	8651
6	526.473	0.32640	165.938	7808
7	466.633	0.28930	139.737	6576
8	397.273	0.24630	119.340	5616
9	334.713	0.20750	100.400	4725
10	274.873	0.17040	74.235	3493
11	231.353	0.14340	63.300	2979
12	194.633	0.12060	48.700	2292
13	182.393	0.11308	42.180	1985
14	140.233	0.08694	34.93	1644

TABLE 3.6.3. RHEOLOGICAL PRPPERTIES

Capillary Tube I.D.= 0.17 cm , L = 67.35 cm

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 2% CMC A Temperature = 33 ⁰ C				
1	1108.719	0.68740	57.618	2711
2	1044.588	0.64764	54.019	2542
3	982.318	0.60903	48.201	2268
4	895.937	0.55547	50.388	2371
5	811.396	0.50306	43.480	2046
6	705.066	0.43714	33.900	1595
7	616.435	0.38218	28.080	1321
8	548.204	0.33988	21.870	1034
9	524.774	0.32535	21.095	993
10	364.343	0.22589	15.200	715
11	275.013	0.17050	10.470	493
12	205.880	0.12764	7.190	338
13	137.690	0.08536	4.100	193
14	104.820	0.06498	3.165	149
15	84.190	0.05219	2.609	123

Contd.....3.6.3

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 2% CMC A Temperature = 41 ⁰ C				
1	1107.419	0.68659	79.606	3746
2	990.228	0.61394	76.360	3593
3	911.34	0.56500	64.400	3030
4	836.080	0.51836	57.389	2701
5	758.300	0.47014	44.485	2093
6	677.800	0.42023	45.630	2147
7	613.715	0.38050	40.878	1924
8	537.324	0.33310	33.320	1568
9	493.574	0.30600	31.020	1459
10	396.783	0.24600	22.120	1041
11	325.832	0.20200	17.058	803
12	263.040	0.16300	13.665	643
13	220.650	0.13670	10.601	499
14	206.820	0.12822	7.562	356
15	124.990	0.07749	5.262	248
16	103.004	0.06386	4.756	224

Contd.....Table 3.6.3

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 2% CMC A		Temperature = 50 ⁰ C		
1	1038.059	0.64359	106.350	5005
2	897.748	0.55660	78.777	3707
3	800.958	0.49659	72.717	3422
4	704.167	0.43658	65.380	3077
5	597.856	0.37067	45.280	2133
6	502.426	0.31150	39.680	1867
7	413.795	0.25650	29.900	1407
8	329.244	0.20413	20.114	947
9	263.734	0.16351	14.290	673
10	177.823	0.11025	10.137	477
11	135.432	0.08396	7.045	332
12	114.800	0.07117	6.300	296
13	84.650	0.05248	3.658	172

Contd....3.6.3

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 2% CMC A Temperature = 60°C				
1	1092.228	0.67700	145.358	6840
2	1010.390	0.62640	126.044	5931
3	902.727	0.55960	103.366	4864
4	792.336	0.49120	93.820	4415
5	703.706	0.43620	86.430	4067
6	619.155	0.38380	68.837	3239
7	522.364	0.32380	59.377	2794
8	435.494	0.27000	44.896	2113
9	341.023	0.21140	35.520	1672
10	271.433	0.16820	23.509	1106
11	206.000	0.12770	17.364	817
12	152.650	0.09460	12.394	583
13	121.370	0.07520	8.980	423

TABLE 3.6.4. RHEOLOGICAL PROPERTIES

Capillary Tube I.D. = 0.17 cm, L = 67.85 cm

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 4% CMC A		Temperature = 40°C		
1	1092.757	0.67750	33.510	1577
2	951.085	0.58967	24.442	1150
3	878.773	0.54483	20.421	961
4	754.781	0.46796	15.721	740
5	649.829	0.40289	13.466	634
6	536.717	0.33276	9.450	445
7	412.725	0.25588	7.000	329
8	337.593	0.20930	4.897	230
9	257.230	0.16940	3.118	147
10	193.069	0.11970	2.105	99
11	150.676	0.09340	1.933	91
12	124.604	0.07720	1.456	69
13	87.949	0.05452	0.958	45

Contd....3.6.4

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 4% CMC A Temperature = 50°C				
1	1037.229	0.64308	36.684	1726
2	948.597	0.58800	38.058	1791
3	853.165	0.52890	30.799	1449
4	756.373	0.46895	26.460	1245
5	679.980	0.42158	21.090	992
6	580.469	0.35989	12.268	813
7	514.957	0.31927	12.672	596
8	415.445	0.25757	10.996	517
9	334.973	0.20768	7.775	365
10	216.421	0.13418	6.830	321
11	194.429	0.12054	5.455	257
12	135.716	0.08414	3.136	158
13	85.164	0.05280	1.460	69

Contd....3.6.4

Run	τ_p	τ_w	U	$\frac{8U}{D_t}$
Fluid= 4% CMC A		Temperature = 60°C		
1	1046.749	0.64898	52.490	2470
2	1002.317	0.62140	47.680	2243
3	917.085	0.56859	47.190	2221
4	802.595	0.49760	37.613	1770
5	697.660	0.43250	27.650	1302
6	596.789	0.37000	25.740	1211
7	485.037	0.30072	19.385	912
8	396.405	0.24577	13.187	621
9	333.600	0.20684	11.257	530
10	266.741	0.16537	8.349	393
11	220.269	0.13656	6.086	286
12	179.236	0.11110	5.270	248
13	140.924	0.08737	3.340	157
14	117.570	0.07289	2.720	128
15	105.100	0.06516	2.355	111

3.7 Experimental Setup

The purpose of the experimental phase of the present investigation is to obtain data for heat transfer to purely viscous non-Newtonian fluids in coil and agitated vessel. The fluids, four aqueous solutions of carboxy methyl cellulose, were selected to meet the following requirements:

- (1) Purely viscous nature of the materials
- (2) Pseudoplastic behavior with varying power law indices and the flow behavior constants.

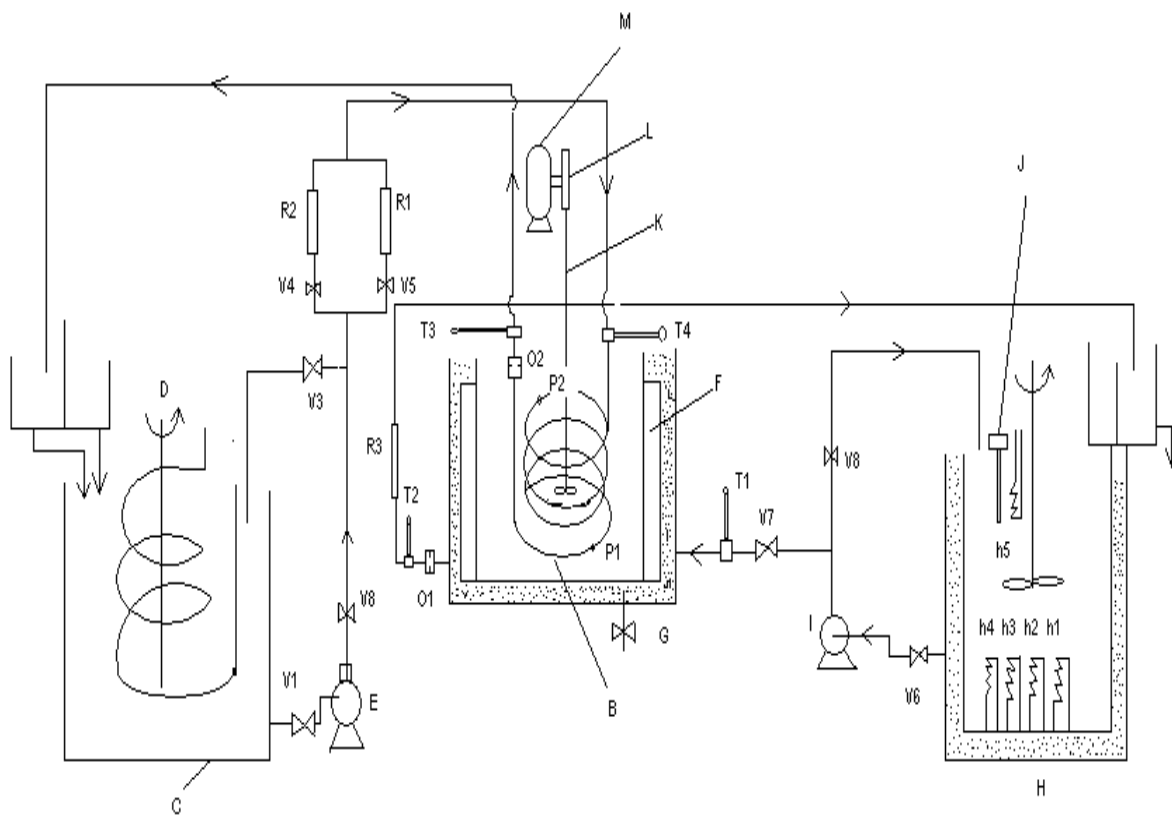
A known weight of solid CMC was dissolved in water to obtain solutions of viscosities in the ranges usually encountered in the industries. A sample of this fluid was collected from the fluid tank during an experimental run and the rheological data were obtained by a capillary tube viscometer.

A schematic diagram of the experimental set up is given in figure 3.7.1.

The set up consisted of a flat bottomed cylindrical test vessel 'A' of 45.25 cm inner diameter and 60 cm height, made from 1/8 inch thick copper sheet. An annular space 'F' around the test vessel was provided by surrounding it with another cylinder of 44 cm height and made from 1/8 inch thick G.I. sheet. The jacket and the vessel assembly was adequately insulated by keeping it in a wooden tank and then filling the surrounding space with glass wool and thermocouple.

A helical coil 'B' having 34 cm mean coil diameter and made from 1.898 cm i.d. and 2.25 cm o.d. copper tubing was placed just in the center of the test vessel. The coil had $5\frac{1}{2}$ turns with 5 cm gap between any two consecutive turns (i.e., $p=5$ cm). About 35 cm tube length at the both the ends of the coil was insulated to prevent heat transfer from entrance and exit regions. The ends of the coil were connected to the two Perspex pipe lengths of 1 inch diameter, one leading to the delivery tank and the other to the pump through rotameters R_1 and R_2 .

In order to measure the pressure drop across the effective length of the coil, pressure tappings t_1 and t_2 were provided by two holes carefully drilled on the surface of the coil. The pressure taps were connected through high pressure polythene tubings to two manometers each of 200 cm



A- TEST VESSEL, B-COIL, C- FLUID STORAGE TANK, D- STIRRER, E- FLUID PUMP, F- JACKET, G- WOODEN TANK WITH INSULATION, I- HOT WATER PUMP, J- TEMPERATURE CONTROLLER, K- AGITATOR, L- REDUCTION GEAR ASSEMBLY, M- MOTOR, O1,O2- ORIFICES, R1,R2,R3- ROTAMETERS, T1,T2,T3,T4- THERMOMETERS, V1,V2,V3,V4,V5,V6,V7,V8- REGULATING VALVES, h1,h2,h3,h4,h5- HEATER, P1,P2- PRESSURE TAPPINGS.

FIG 3.7.A SCHEMATIC DIAGRAM OF THE EXPERIMENTAL SET UP



FIG 3.7.B PHOTOGRAPH EXPERIMENTAL SET UP



FIG 3.7.C PHOTOGRAPH OF EXPERIMENTAL SET UP

height and containing fluids of different densities (CCL_4 and Hg). The former measured lower pressure drops while the latter is used to measure higher pressure gradients.

The fluid which circulated through the coil was contained in a delivery tank 'C' of 200 liters capacity and was provided with a marine type agitator 'D'. The fluid was circulated through the coil by a 5 H.P., 3 stage centrifugal pump 'E' through the rotameters R_1 and R_2 of measuring capacity 500 and 1000 gph, respectively, connected in parallel. A bypass at the pump discharge side and the regulating valves V_1 , V_2 , V_3 , V_4 and V_5 were used to control the rate of flow through the coil.

A rectangular tank of about 300 liters capacity, fitted with five immersion heaters h_1 , h_2 , h_3 , h_4 and h_5 of 2000 watts each, was used to heat the water which in turn was circulated through the jacket (annular space around the test vessel) with the help of a centrifugal pump I. The temperature of water in the tank was maintained constant with the help of a mercury-toluene temperature control device 'j' connected in the circuit of the control heater.

The fluid in the test vessel was agitated by marine type agitator 'K' fitted in the centre of the coil. The agitator shaft was driven at a known speed by a 2 H.P. electric motor 'M' through a reduction gear assembly 'L'. Provision was made at the bottom end of the agitator shaft to replace the impellers of required size and shapes.

Orifices O_1 and O_2 of $\frac{1}{2}$ inch diameter were fixed at the inlet and exit ends of the coil as well as the jacket for proper mixing of the fluids. Thermometers T_1 , T_2 , T_3 , T_4 of 0.1°C calibration were inserted at approximately $\frac{1}{2}$ inch distance on the downstream side of these orifices to measure the inlet and exit bulk temperature.

The mass rates of flow of the fluids through the coil as well as the jacket were measured by collecting the fluids in measuring vessels for a known interval of time and then accurately weighing them.

3.8 PROCEDURE

Before starting the heat transfer measurements, the coil and the jacketed vessel were washed with dilute (1%) HCL for about 20 minutes, and this was followed by a water wash. For measurements with water, the two storage tanks and the test vessel were filled with water. The water level in the test vessel was maintained at a known height of 44 cm. This water level in the

vessel was such that the coil and the agitator blades were completely submerged into it. The height of the blade assembly from the bottom of the vessel was measured accurately.

The water in the hot water tank was heated and the temperature of the fluid was brought to the desired level. The temperature was controlled with the help of a temperature controller and was maintained at a desired value within ± 0.1 °C. Both the pumps were then started and the flow rates in the coil as well as in the jacket were adjusted by means of regulating valves and the bypasses. The agitator rotation was then started at a fixed rpm. When the experiment was in progress, steady state was considered to reach when both inlet and exit-fluid temperatures were essentially constant for a 20 minutes interval. The following observations were recorded.

Inlet and exit fluid temperatures in the jacket and the coil, amount of the water from the jacket and the coil collected separately for a known time interval, rpm of the agitator, temperature of the fluid in the agitated vessel and that in the storage tanks.

The readings were duplicated to ensure the steady state and to eliminate any error in the measurement. Similar measurements were made by varying the flow rate in the coil and then by varying the rotational speed of the agitator.

The above procedure was repeated with all the non-Newtonian fluids for different agitational speeds and coil flow rates. Adequate care was taken to maintain constant temperatures in both the storage tanks during a particular run period.

The following sets of the experiments were conducted:

1. In order to check the accuracy of the experimental set up, data were taken on a Newtonian fluids for different flow rates and the agitational speeds.
2. In order to propose the laminar and turbulent heat transfer correlations for the flow of power law non-Newtonian fluids in coils, the data were taken with four aqueous CMC solutions at different flow rates.
3. Data were taken with different agitational speeds to propose correlations for jacket and coil outside heat transfer.
4. Pressure drops were measured during each coil flow rate studied, to present a correlation for momentum transfer to non-Newtonian fluids flowing through the coil.

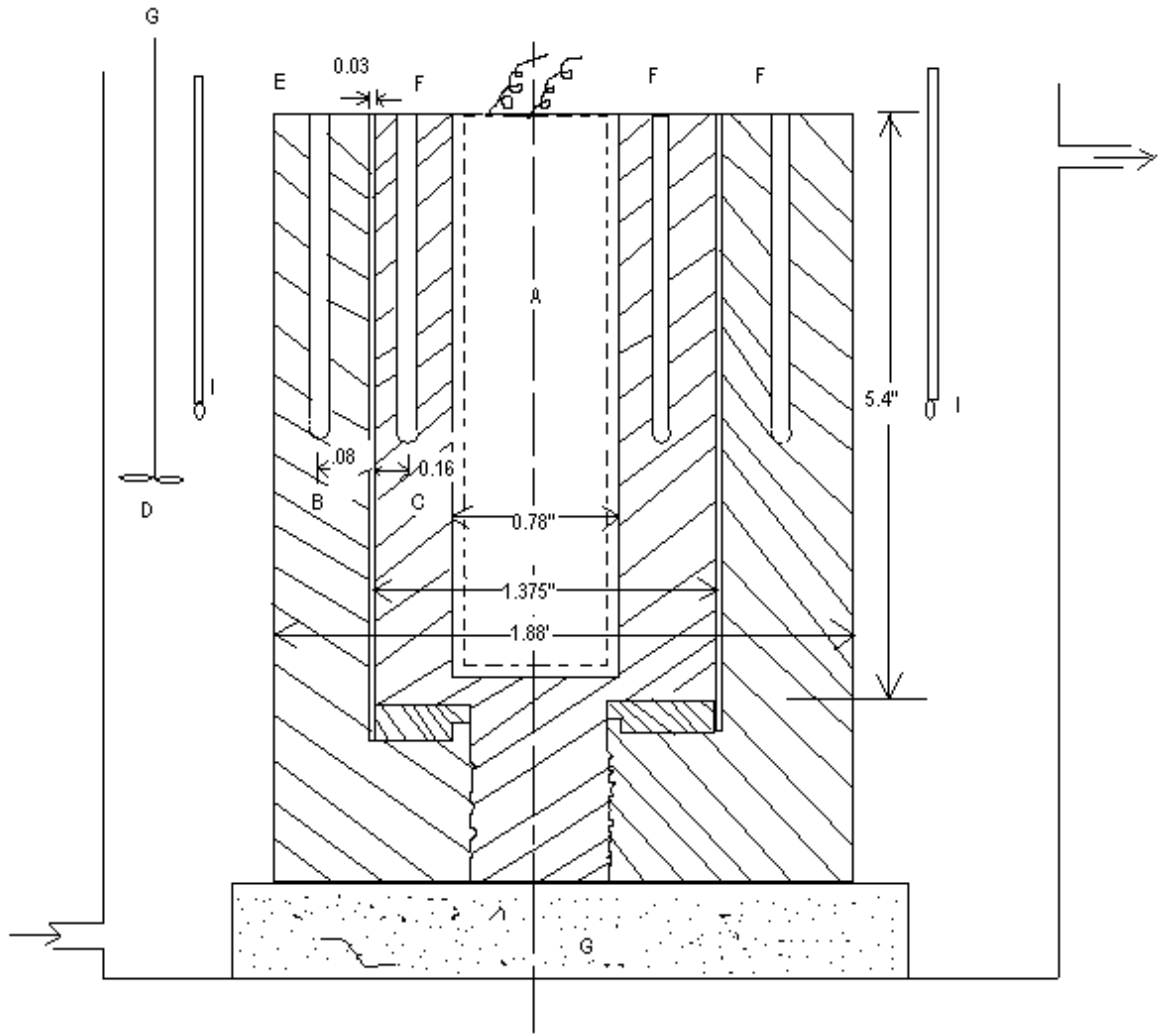
3.9 DETERMINATION OF THERMAL CONDUCTIVITY

Metzner (1961) has concluded that thermal conductivities of non-Newtonian materials should be evaluated experimentally unless all components separately exhibit substantially identical conductivities or, as in the case of suspensions of discrete particles, Maxwell equation or one of their modifications may be used. Further, presently available data suggest the absence of measurable effect of fluid deformation rate upon thermal conductivity.

Thermal conductivities of dilute polymer solutions have been measured by a number of investigators and were generally found to be indistinguishable from those of the solvent at the same temperature level and in fact were not considered further in the most of the recent publications. Thermal conductivity is a more difficult property to measure because of the need to eliminate convective motion within the fluid, and experimental errors like heat losses, heat transfer by radiation, and eccentricity between the internal and external cylinders. Therefore, the 'relative concentric cylinder method' has been used to determine the thermal conductivity of the non-Newtonian fluids. The basic method of measuring the numerical values of thermal conductivity is based on the Fourier's law.

3.9.1 APPARATUS

The schematic diagram of the apparatus is shown in figure 3.9.1. It consists of two copper concentric cylinders with the annular space of 0.067 cm width and 13.22 cm length. The outer diameter of the inner cylinder is being 3.51 cm. two holes each of 1.58 mm diameter and 5.08 cm long are drilled in the wall of each cylinder providing the space for the copper-constantan thermocouples used for measuring the wall temperatures. 1.99 cm diameter bore in the centre of the inner cylinder provides the space for the electrical heater. Insulators 'H' and 'G' minimize the heat losses in longitudinal directions.



- A. HEATER
- B. OUTER CYLINDER
- C. INNER CYLINDER
- D. WATER BATH
- E. THERMOCOUPLES
- F. THERMOCOUPLES
- G. INSULATION
- H. INSULATION

FIG: 3.9.1 APPARATUS FOR THERMAL CONDUCTIVITY MEASUREMENT

3.9.2 PROCEDURE

Length and inner and outer diameters of both the cylinders were measured. The test fluid was filled in the groove of the outer cylinder, which filled the annular space when the inner cylinder was inserted and screwed in the outer cylinder. Some more test fluid was injected in the annular space from the top till it filled the space completely and formed an annular liquid film between the two cylinders. Four copper-constantan thermocouples were placed in the holes meant for the measurement of the cylinder wall temperatures. This set of the apparatus was then inserted in the thermostatic bath. The heater was kept on to the desired voltage and power supply. The inner cylinder was heated uniformly until steady state was established. Under these conditions, a heat flux passing through the cylinder walls and the liquid film would change the temperature in the radial direction. After two or three hours of the attainment of thermal equilibrium, the following measurements were taken: electrical current and voltage (or power supply directly from the wattmeter), the temperature of the thermostatic bath and the emf generated by thermocouples. The temperature of the wall was obtained from the emf-temperature chart prepared from the calibration of the thermocouples.

For steady state conditions the amount of heat transferred per unit time is given by,

$$Q - Q_{loss} = \frac{\Delta t}{\frac{\Delta X_1}{K_{m1} A_{m1}} + \frac{\Delta X}{K_m A_m} + \frac{\Delta X_2}{K_{m2} A_{m2}}} \quad (3.9.2.1)$$

Where Q = total rate of heat input by the heater,

Q_{loss} = heat loss due to the radiation etc.,

ΔX_1 = thickness between the thermocouples and the inner surface of the outer cylinder,

ΔX_2 = thickness between the thermocouples and the outer surface of the liquid film,

ΔX = thickness of the liquid film,

Δt = temperature difference recorded by the thermocouples.

The above equation may be written as follows:

$$Q - Q_{loss} = \frac{\Delta t}{\frac{A+B}{K_m}} \quad (3.9.2.2)$$

Where A and B are constants. Constants A and B were calculated and found to be $A = 0.000580$ and $B = 0.01336$. Assuming Q_{loss} to be the same for equal temperature range, the numerical values of A and B were verified by the standard liquids, 'water' and 'ethyl alcohol' of known thermal conductivities (T6). The thickness of the annular film formed was verified by the measurements on carbon tetrachloride. Thermal conductivities of the solutions were calculated by comparing the heat fluxes for solution and solvents keeping the same range of temperature difference or keeping the heat losses, Q_{loss} , the same.

3.10 DETERMINATION OF SPECIFIC HEAT

Specific heat of the solvent and the solution were measured by calorimetric method and were found to be nearly equal to that of water.

Specific heat measurement by blending (mixing) The sample material characterized by known weight m , temperature t and unknown specific heat is mixed (blended) with a material with known parameters Specific heat, mass, temperature - mostly with a liquid usually not chemically reacting with the sample material to be measured. We measure the common equilibrium temperature. Provided that during mixing only heat transfer occurs between the two materials and there is no heat loss, the heat taken up (or emitted) by the test sample is equal to the heat emitted (or taken up) by the other material with known parameter, therefore on the basis of which the unknown heat capacity (c) can be determined. This method is primarily used for the specific heat determination of solids and liquids, but applicable also to constant pressure heat capacity measurement of gases. In this case, the sample gas is flowing in a spiral tube through the liquid. Knowing the mass and the temperature change of the gas, as well as all the data of the measurement liquid, the constant pressure specific heat of the gas (c_p) can be calculated.

With the possession of the heat capacity at constant pressure, the constant volume heat capacity (c_v) of the gas can also be determined.

CHAPTER 4

4.1 RESULT AND DISCUSSION

The results of heat transfer to non-Newtonian fluids in agitated vessel and in the helical coil are discussed in the following pages. In order to interpret the results clearly, the entire accumulated experimental data are discussed in three sections. First part is concerned with the heat transfer from the vessel wall to the agitated fluids; the second part deals with the heat transfer from agitated fluids to the coil while in the third section the heat transfer results of the fluids flowing inside the coiled pipe are discussed. Correlations of the experimental results are based on the theoretical background presented in chapter 3. The heat transfer characteristics of the five fluids, viz., water and four aqueous CMC-A solutions of concentrations 0.5, 1, 2 and 4% by weight have been investigated. The method of determination of their rheological properties is described in chapter 3 and the viscometric data are given in table 3.6.1 through 3.6.4. All the CMC solutions were considered as pseudo plastic fluids obeying power law relation

$$\tau = K \left(-\frac{du}{dr} \right)^n \quad (4.1)$$

Behavior of these solutions and their shear diagrams are shown in figure 3.5.4.A through 3.5.4.D. The flow behavior index, 'n', at various temperatures was found to be the same for a particular fluid whereas the consistency index, K, decreases with the temperature as shown in figure 3.5.4.E. Three aqueous solutions containing 0.5 (n=0.973), 1 (n= 0.851) and 2% (n=0.793) CMC-A and water were studied in the coil and 1, 2 and 4% (n=0.698) aqueous solutions of CMC-A in the vessel. The thermal conductivities of CMC-A solutions were determined by comparative concentric cylinders method described in chapter V and were found to be very near or equal to that of water. The specific heats of these fluids were also found to be equal to that of water and no appreciable change was seen due to varying polymeric concentration in the water.

4.1.1 Range of Variables

Agitation was provided separately by three, 4 bladed marine types of agitators of diameters 7.5, 12.7 and 18.35 cm. vessel, coil and coil tube diameters were not varied and were 45.25, 34.0 and 1.898 cm, respectively. Flow behavior index, n, of the fluids in the vessel ranged from 0.69 to 1

and of the fluids in the coil from 0.79 to 1. A detailed list of the range of various dimensionless groups is given in tables 5.1.A and 5.1.B.

4.1.2 Heat Balance and Overall Heat Transfer Coefficient

In the heat balance the heat produced due to mechanical agitation and heat losses due to evaporation, radiation and natural convection were considered to be negligible and further the two effects tend to compensate each other. Overall heat transfer coefficients, U_j , and U_{oc} , for jacketed vessel wall (based on inside surface of the vessel) and for helical coil (based on outside surface of the coil tube), respectively, were calculated from the following overall heat balance equations under the steady state conditions:

$$Q = W_j C_{Pj} (t_{j1} - t_{jo}) = U_j A_j (t_j - T)_{1m} \quad (4.1.2.1)$$

$$Q = W_c C_{Pc} (t_{co} - t_{ci}) = U_{oc} A_{oc} (T - t_c)_{1m} \quad (4.1.2.2)$$

4.1.3 Analytical Procedure for Individual Heat Transfer Coefficients, h_j , h_{oc} and h_{ic}

Liquid film heat transfer coefficients h_j were calculated from the overall coefficient U_j using modified Wilson graphical method. The overall resistance $1/u_j$ is equal to the liquid film resistance plus jacket side resistance including that of the vessel wall. The two coefficients can be related by

$$\frac{1}{U_j} = \frac{1}{h_j} + Y_j \quad (4.1.3.1)$$

Many of the previous workers have found that for Newtonian fluids

$$h_j = \phi_1 N_{Re}^{2/3} N_{Pr}^{1/3} \quad (4.1.3.2)$$

Or

$$h_j = \phi_2 N^{2/3} \quad (4.1.3.3)$$

Thus, for these fluids a plot of $\frac{1}{U_j}$ versus $\frac{1}{N^{2/3}}$ will yield a straight line with intercept Y_j and this makes the calculation of individual h_j possible. However, this analysis fails in the case of non-Newtonian fluids because of the following reasons. Conventional 2/3 and 1/3 exponents of Reynolds and Prandtl groups, respectively, are found to be equal for both Newtonian and non-Newtonian fluids. A closer observation of various Reynolds and Prandtl groups defined by

Metzner and Otto (1957), Calderbank and Moo Young (1957), Gluz and Pevlushenko (1966) and Hagedorn and Salamone (1967), working on pseudo plastic fluids, reveals that

$$N_{Re} \propto N^{2-n} \quad (4.1.3.4)$$

and

$$N_{Pr} \propto N^{n-1} \quad (4.1.3.5)$$

Thus, with the help of Equations (4.1.3.4) through (4.1.3.5), it may easily be written that

$$h_j = \psi (N^{2-n})^{2/3} (N^{n-1})^{1/3}$$

Or

$$h_j = \psi_1 N^{(3-n)/3} \quad (4.1.3.6)$$

For Newtonian fluids equations (4.1.3.3) and (4.1.3.6) are one and the same. Substitution of equation (4.1.3.6) in equation (4.1.3.1) gives

$$\frac{1}{U_j} = \frac{1}{\psi_1 N^{(3-n)/3}} + Y_j \quad (4.1.3.7)$$

Equation (4.1.3.7) shows that for power law fluids a modified Wilson plot of $1/U_j$ versus

$\frac{1}{N^{(3-n)/3}}$ will be more appropriate than that of $1/U_j$ versus $\left(\frac{1}{N_{Re}^{2/3}}\right)$ or of $1/U_j$ versus

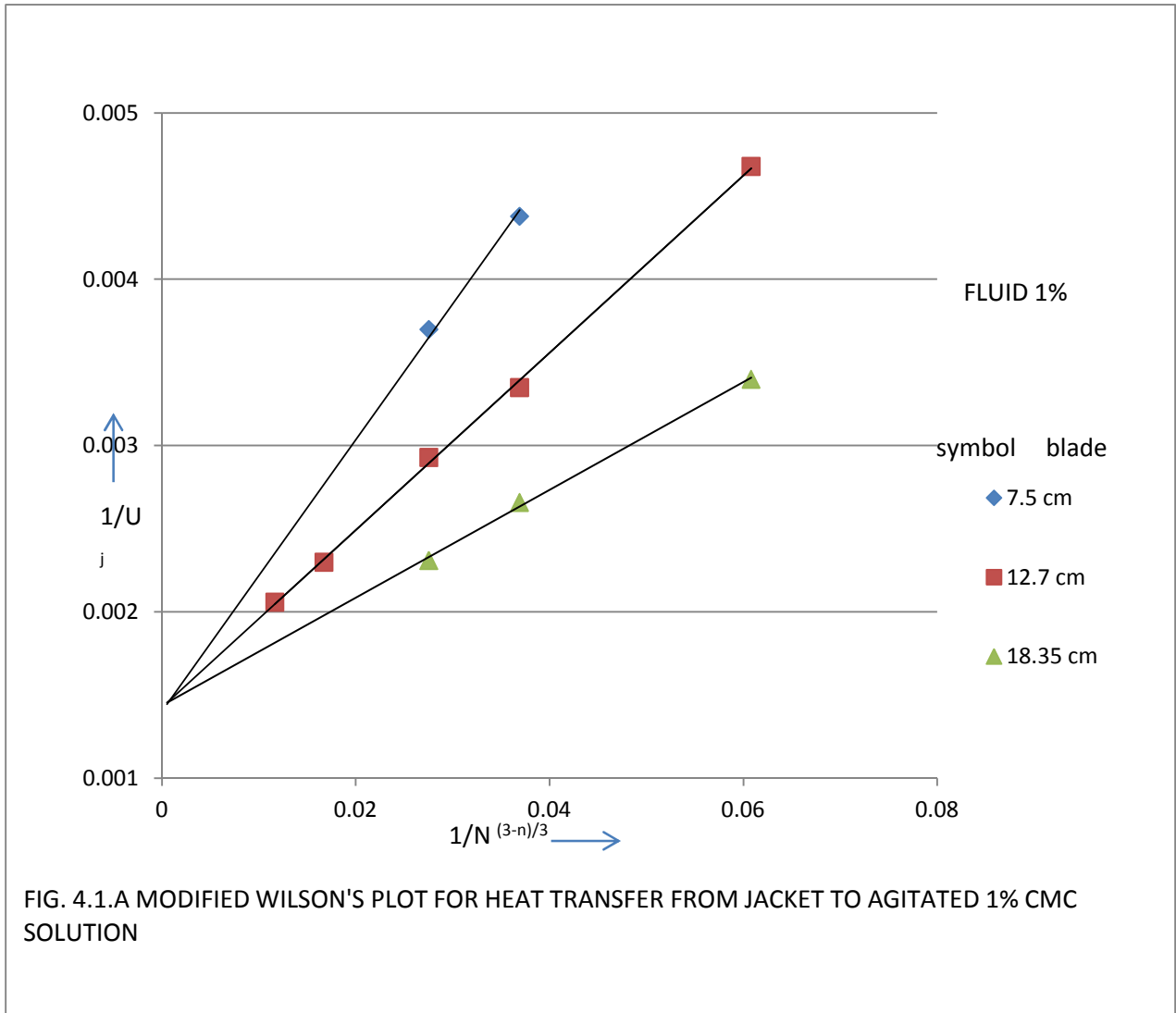
$\frac{1}{N^{2(2-n)/3}}$ used by Carreau et al. (1959). In figure 4.1.A, $1/U_j$ is plotted against $\frac{1}{N^{2/3}}$ for the

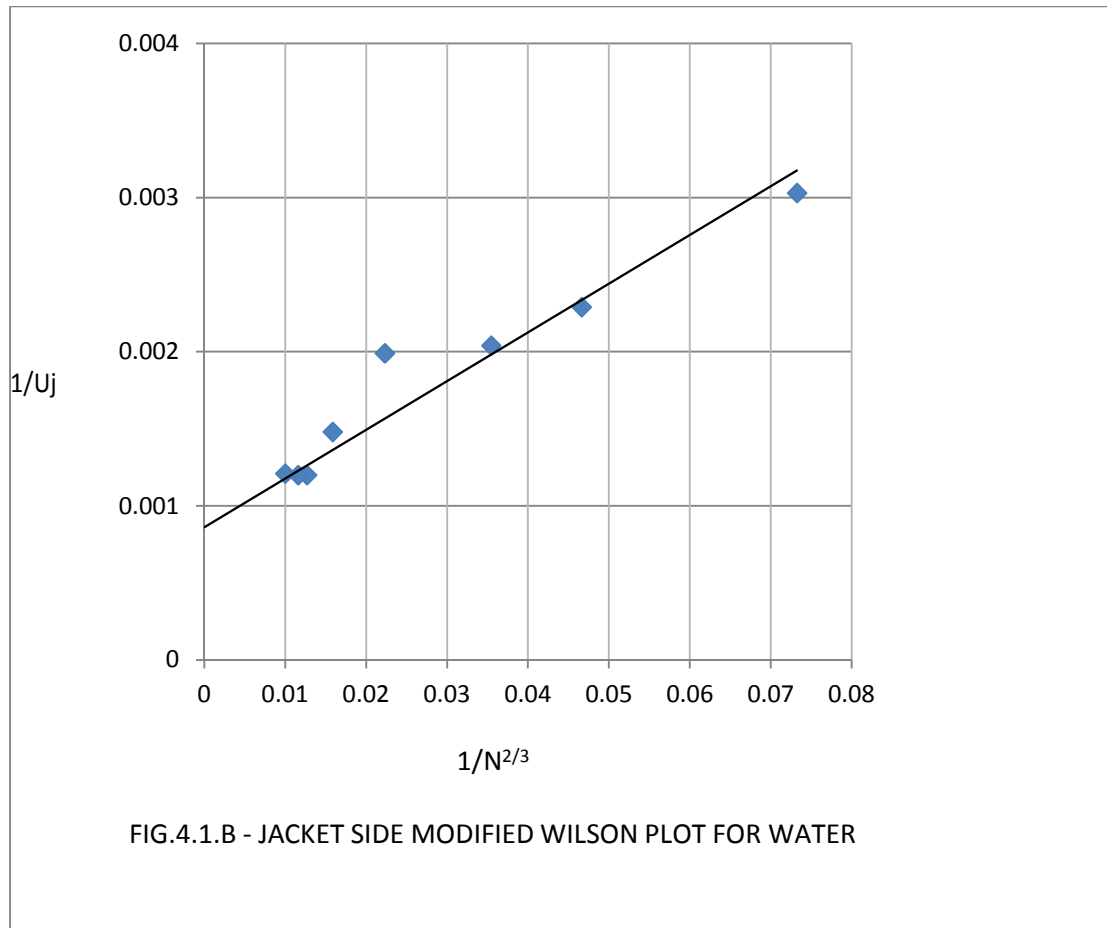
case when the vessel fluid was water. Y_j in this case is found to be 0.00086. $1/U_j$ is plotted

against $\frac{1}{N^{(3-n)/3}}$ in figure 4.1.B, through 4.1.D for 1, 2 and 4% CMC-A solutions,

respectively. An interesting feature of these plots is that for any vessel fluid the average lines drawn through the data points for different blade sizes converge at the same point on the $1/U_j$ ordinate. This strengthens the utility of present modification of the Wilson plot as shown by equation (4.1.3.7). An additional average of this plot is that it gives a very clear picture of the effect of the impeller blade diameter on the overall heat transfer coefficient. The values of Y_j are found to be 0.00145, 0.00178 and 0.0012 for 1, 2 and 4% CMC-A studied in sequence. The increase in the constant Y_j is due to gradual deposition of a thin layer of dirt on the jacket side

surface of the vessel wall and also due to slight variation in the jacket side fluid flow rate, although it was maintained constant during runs with a particular fluid in the vessel.





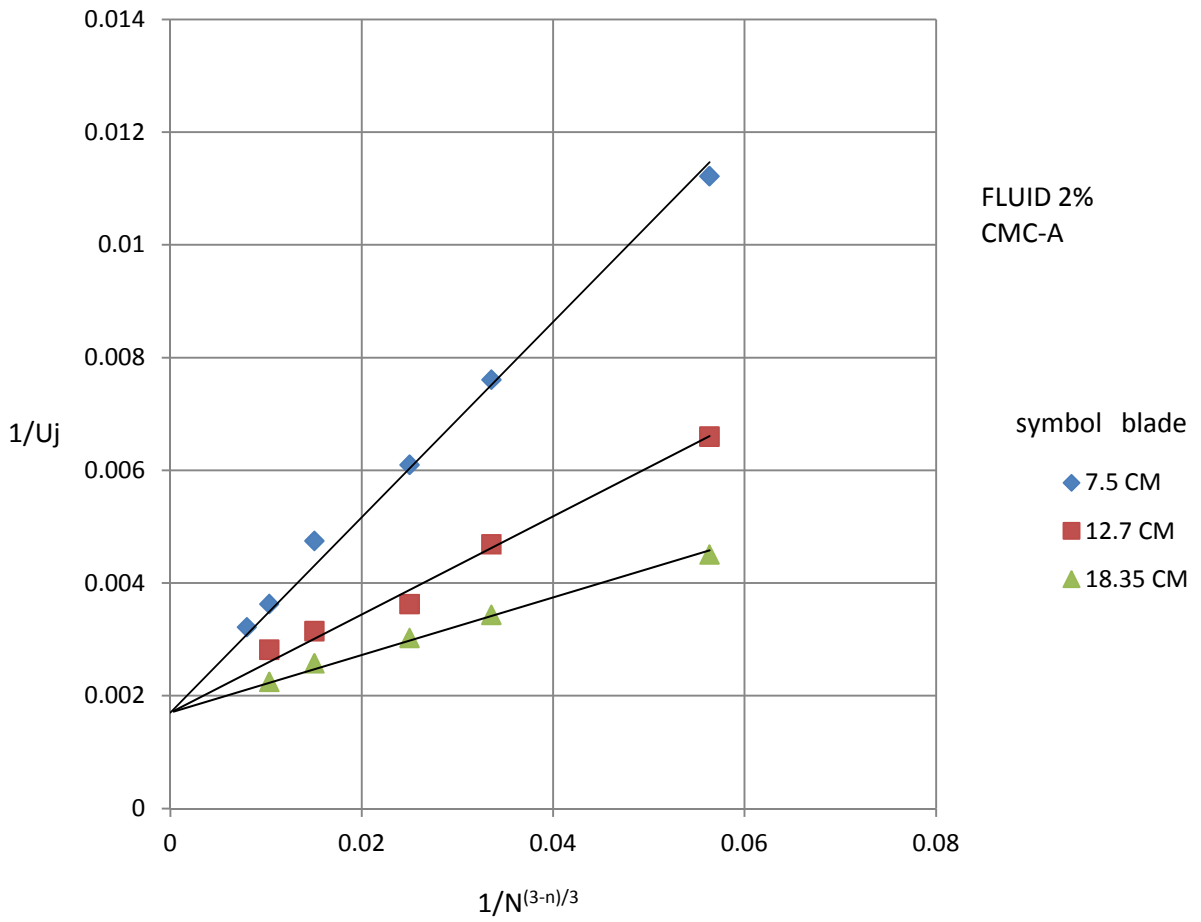
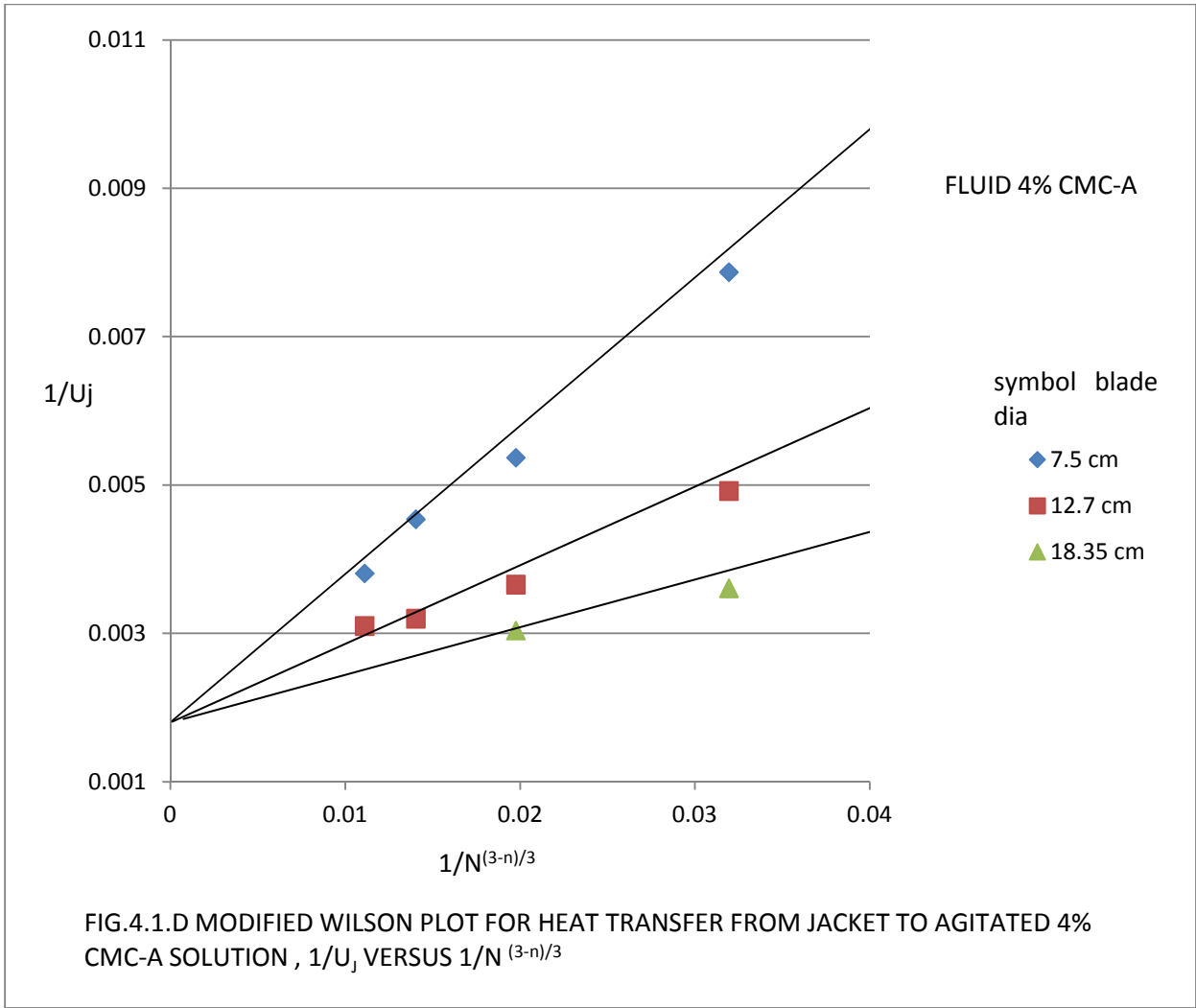


FIG. 4.1.C- MODIFIED WILSON'S PLOT FOR HEAT TRANSFER JACKET TO AGITATED 2% CMC -A SOLUTIONS



The above procedure was also followed for the determination of h_{oc} , the outside heat transfer coefficient of the coil tube from the coil side overall heat transfer coefficient, U_{oc} given by

$$\frac{1}{U_{oc}} = \frac{1}{h_{oc}} + Y_c \quad (4.1.3.8)$$

Where,

$$Y_c = \frac{1}{h_{ic}} \frac{D_o}{D_t} + \frac{\Delta X D_o}{k_m D_{tm}} + \text{dirt resistance} \quad (4.1.3.9)$$

And

$$h_{oc} = \psi_c N^{(3-n)/3} \quad (4.1.3.10)$$

$1/U_{oc}$ versus $1/N^{2/3}$ plot for water as shown in figure 4.2.F gives Y_c equal to 0.00012. Further, figure 4.2.G (vessel fluid: 4% CMC-A and coil fluid: 0.5% CMC-A flowing at constant flow rate), 4.2.H (vessel fluid: 1% CMC-A and coil fluid: 1% CMC-A flowing at constant rate), 4.2.I (vessel fluid: 2% CMC-A and coil fluid: 2% CMC-A at constant rate) where $1/U_{oc}$ is plotted against $1/N^{(3-n)/3}$, also yield separate straight lines for different blade diameters. The respective

Y_c values are found to be 0.000445, 0.000575 and 0.000905. Calculated variables $1/U_j$, $1/U_{oc}$, N , h_j and h_{oc} for various fluids and blades studied in the present work are listed in tables 7.2 through 7.5.

The liquid film coefficient inside the coil tube i.e. h_{ic} was calculated by the relation

$$\frac{1}{h_{ic}} \left(\frac{1}{U_{oc}} - \frac{1}{h_{oc}} \right) \frac{D_t}{D_o} \quad (4.1.3.11)$$

For a set of observations the fluid flow rate in the coil was varied at a constant rotational speed of the agitator (say at N') and thus for that set of reading h'_{oc} was maintained constant. The value of h'_{oc} was obtained from Wilson plot according to the relation

$$\frac{1}{h'_{oc}} = \frac{1}{U_{oc}} - Y_c \quad (4.1.3.12)$$

Calculated values of h_{ic} and other experimental data are presented in table 5.16 through 5.19 for water, and 0.5, 1 and 2% CMC-A, respectively.

4.1.4 Non Isothermal Correction

The temperatures of the vessel and the tube walls were calculated by the known overall and individual film resistances. The difference between the temperatures of the wall and the fluid was found to range from 1 to 6⁰ C. For non-Newtonian fluids, the ratio of the viscosities evaluated at wall and fluid bulk temperatures could be approximated by the ratio of corresponding consistency indices K_w and K_b . The ratio $\left(K_b/K_w\right)$ was found to vary from 1.02 to 1.08. Thus, the correction factor $(K_w/K_b)^a$, used by previous workers with the exponent, a , varying from 0.12 to 0.24, becomes nearly equal to 1. Further, the operation could be assumed to be isothermal one and therefore non-isothermal correction factor is not used in the correlation presented in the later sections.

4.2 CORRELATIONS

4.2.1 Heat Transfer from Jacket to Agitated Non-Newtonian Fluids

In order to evaluate constant C_1 and Reynolds and Prandtl number indices b_1 and b_2 , respectively, in equation (3.3.24), viscosity μ_a'' and dimensionless groups were calculated using the following equations discussed in chapter 3.

$$N_{Nuj} = \frac{h_j D_T}{k}$$

$$\mu_a'' = n^{1-n} K (4\pi N)^{n-1}$$

Reynolds and the Prandtl number may be written as

$$N_{Rea}'' = \frac{D_a^n (\pi D_a N)^{2-n} \rho}{K n^{1-n} 4^{n-1}} \quad (4.2.1.1)$$

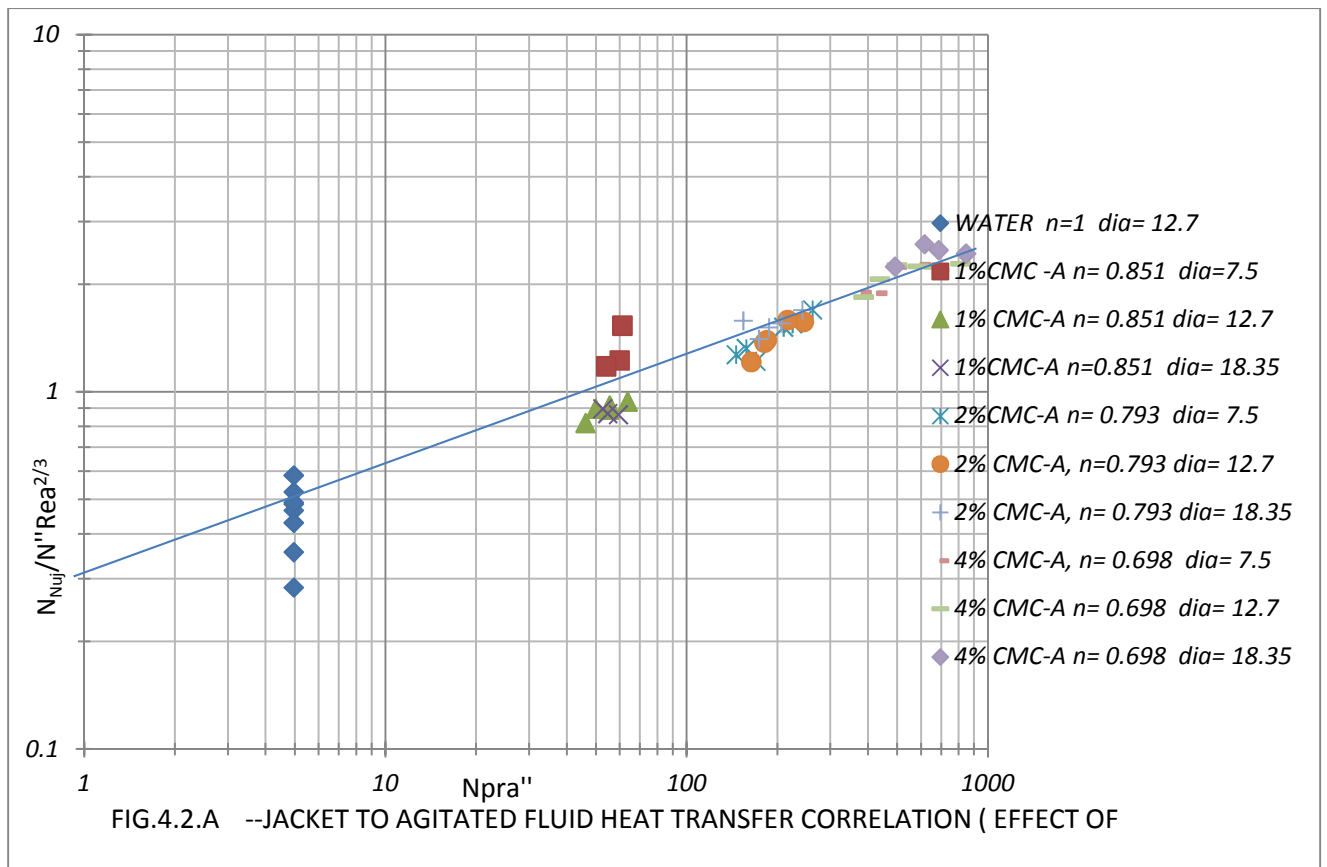
$$N_{Pra}'' = \frac{C_p K (4\pi N/n)^{n-1}}{k} \quad (4.2.1.2)$$

Calculated values of N_{Nuj} , N_{Rea}'' , N_{Pra}'' are given in tables 5.6 through 5.9 for water and 1, 2 and 4% CMC-A, respectively.

4.2.2 Effect of Prandtl Number

Prandtl number influence is shown in figure 4.2.A where $N_{Nuj}''/N_{Rea}''^{2/3}$ is plotted against N_{Pra}'' and the influence of Reynolds number is eliminated by dividing N_{Nuj}'' values by $N_{Rea}''^{2/3}$.

The exponent of N_{Rea}'' was chosen as 2/3 from the previous experience. Prandtl number of polymeric solutions obeying power law varies with the speed of the agitator and is found to vary from a minimum of 4.9 for water to a maximum of 850 for 4% CMC solution in the present investigation. The average line through the data points of figure1 gave the exponent b_2 of the Prandtl number equal to 1/3 which agrees with Prandtl number exponent in heat transfer correlation for Newtonian fluids.



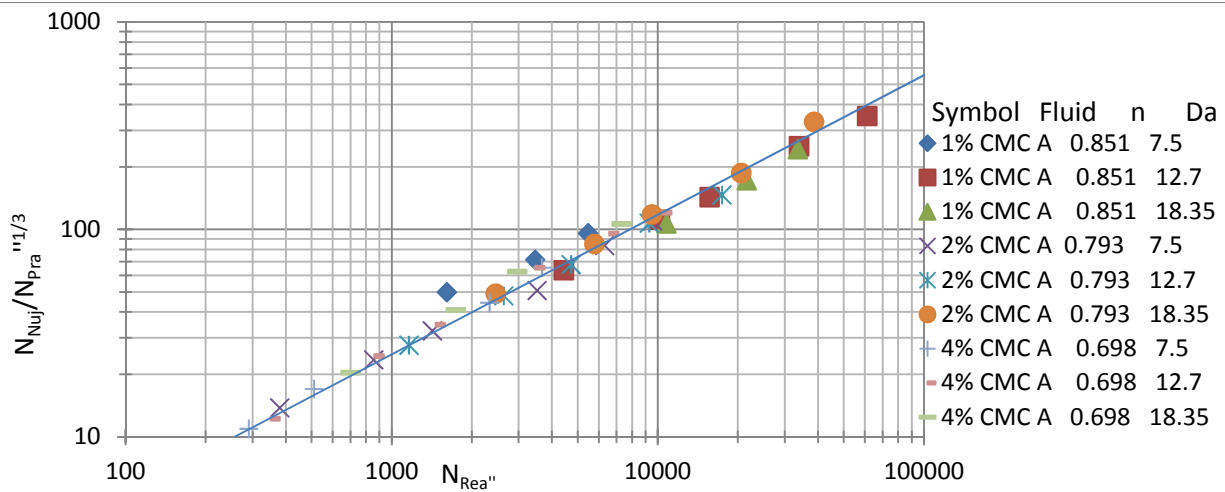


FIG.4.2.B: HEAT TRANSFER CORELLATION FOR AGITATED NON-NEWTONIAN FLUIDS (EFFECT OF REYNOLDS NUMBER)

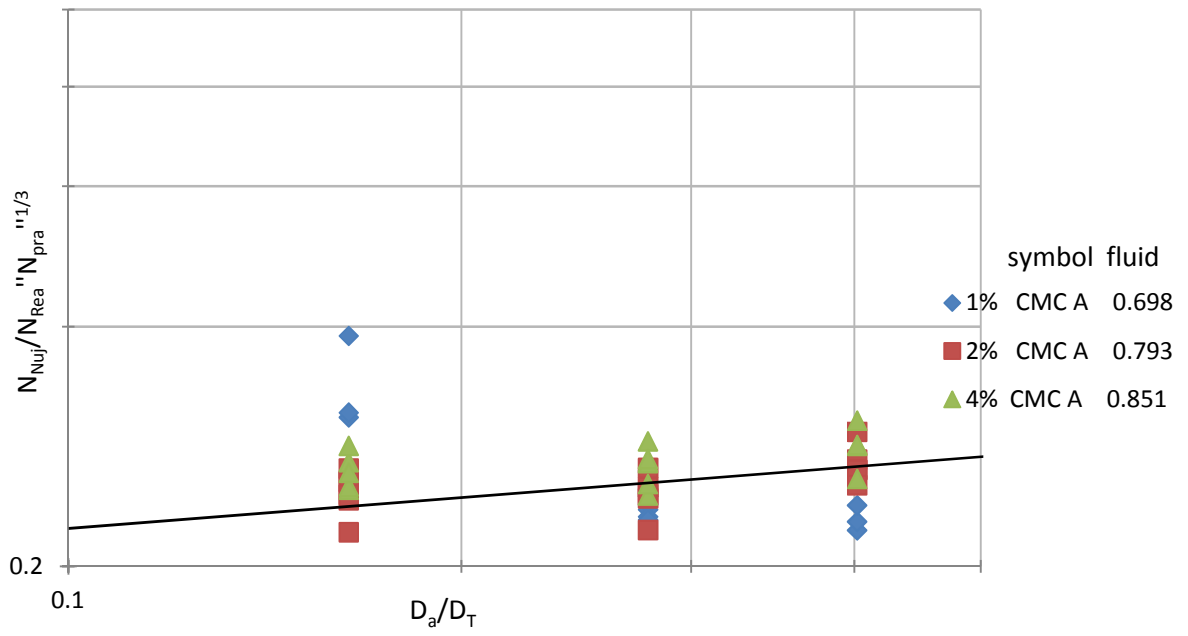


FIG 4.2.C.- HEAT TRANSFER CORRELATION FOR JACKET TO AGITATED FLUIDS:
EFFECT OF AGITATOR DIAMETER.

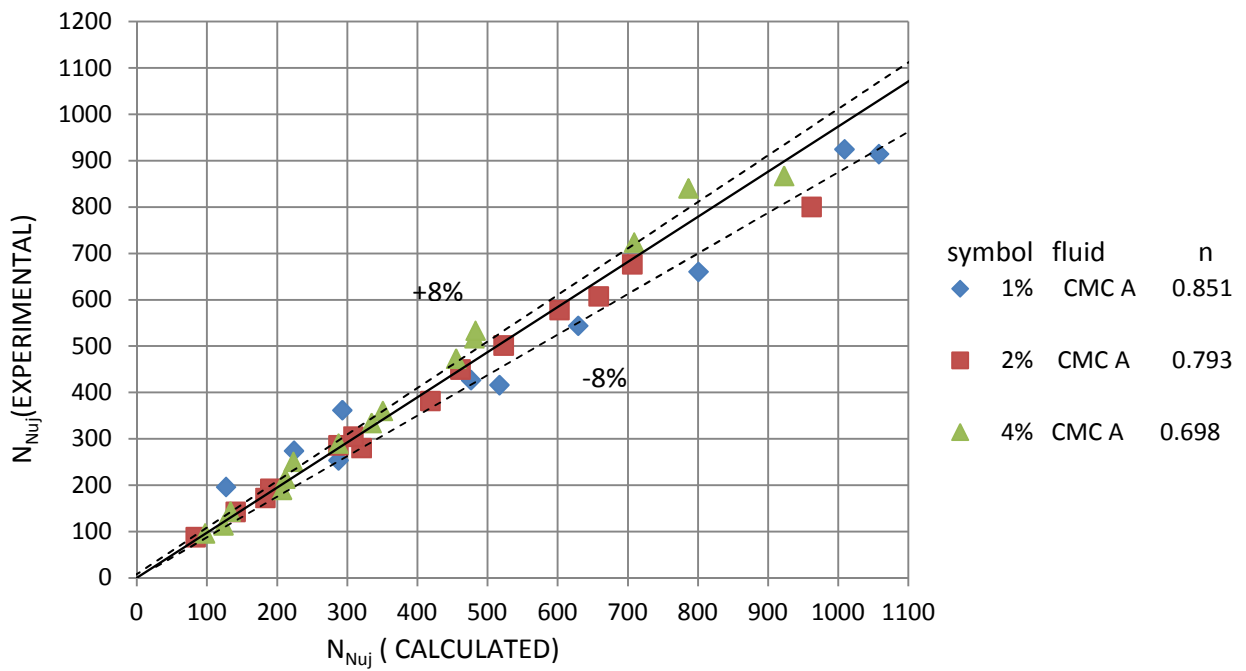


FIG4.2.E - COMPARISON BETWEEN EXPERIMENTAL AND CALCULATED VALUES OF NUSSELT NUMBER N_{Nu_j} (HEAT TRANSFER FROM JACKET TO AGITATED FLUIDS)

4.2.3 Effect of Reynolds Number

A logarithmic plot of $N_{Nuj}/N_{Pra}''^{1/3}$ versus N_{Rea}'' for three fluids (1, 2 and 4% CMC-A) is shown in figure 4.2.B. This plot gave an average line with a slope equal to $2/3$. Thus the validity of the assumed exponent of $2/3$, used in figure 4.2.A and is modified Wilson plots is proved. The average line of slope (the exponent b_1 of Reynolds number in equation 3.3.24 drawn in figure 4.2.B also agrees well with previous exponent of Reynolds number in the correlations for Newtonian fluids.

4.2.4 Effect of Impeller diameter

For studying the effect of impeller diameter, the heat transfer measurements were made with three impeller of diameter 7.5, 12.7 and 18.35 cm respectively. Although the diameter of impeller, D_a shows its effect is Reynolds number, an attempt has been made to find the effect of D_a/D_T ratio on Nusselt number. Figure 4.2.C shows a correlation between $(N_{uj}/N_{Pra}''^{1/3} N_{Rea}''^{2/3})$ and D_a/D_T in which data of three fluids for D_a/D_T ratio 0.166, 0.282 and 0.403 have been plotted. Almost negligible effect of D_a/D_T is observed possibly because of very small range of D_a/D_T ratio covered in the present work. However, a mean line through data point shows the index of D_a/D_T equal to 0.1. Thus the overall effect of D_a , including its presence in Reynolds number, gives

$$h_j \propto D_a^{1.433} \quad (4.2.4.1)$$

Finally $N_{uj}/N_{Pra}''^{1/3} (D_a/D_T)^{0.1}$ is plotted against Reynolds number N_{Rea}'' in figure 4.2.D which includes data for water and 1,2 and 4% CMC solution. The method of least square gives the equatin

$$N_{Nuj} = 0.302 N_{Rea}''^{2/3} N_{Pra}''^{1/3} \left(\frac{D_a}{D_T}\right)^{0.1} \quad (4.2.4.2)$$

It is interesting to note that all data of both Newtonian and non-Newtonian fluids are in excellent agreement with equation 4.2.4.1 for $290 < N_{Rea}'' < 1.4 \times 10^7$; $4.9 < N_{Pra}'' < 850$ and $0.166 < (D_a/D_T) < 0.403$. Figure 4.2.E compares the experimental Nusselt numbers with those calculated from equation 4.2.4.2. The standard deviation of 50 data points from equation 4.2.4.2 is found to be 8.03%.

4.2.5 Heat Transfer from Agitated Non-Newtonian Fluids to the Helical Coil

A single copper helical coil ($D_t = 1.898$ cm, $D_c = 34$ cm and of $5\frac{1}{2}$ turns) was studied with water as Newtonian and 1, 2 and 4% CMC-A as three power law non-Newtonian fluids. The agitation was provided by three impellers as discussed previously. Nusselt number, N''_{Rea} , defined by equation (4.2.1.1), Prandtl number defined by equation (4.1.1.2) and other calculated parameters are presented in tables 5.11 through 5.14.

The first objective of this part of investigation is to determine Reynolds and Prandtl number indices b_3 and b_4 , respectively, in equation (3.4.2). Various logarithmic plots and method of correlation are similar to those presented in the previous section of this chapter. As shown in figure 4.2.J, the plot of $N_{Nuoc}/N''_{Rea}{}^{2/3}$ versus N''_{Pra} resulted in a straight line, the slope of which gave the exponent, b_4 , equal to $1/3$. By cross-plotting these data points as shown in figure 4.2.K i.e. as $N_{Nuoc}/N''_{Pra}{}^{1/3}$ versus N''_{Rea} , the exponent of Reynolds number was found to be $2/3$. Further, $N_{Nuoc}/N''_{Rea}{}^{2/3} N''_{Pra}{}^{1/3}$ when plotted against (D_a/D_c) as shown in figure 4.2.L gave the exponent of (D_a/D_c) as 0.10. Thus figures 4.2.J, 4.2.K and 4.2.L resulted in the correlation of the following form:

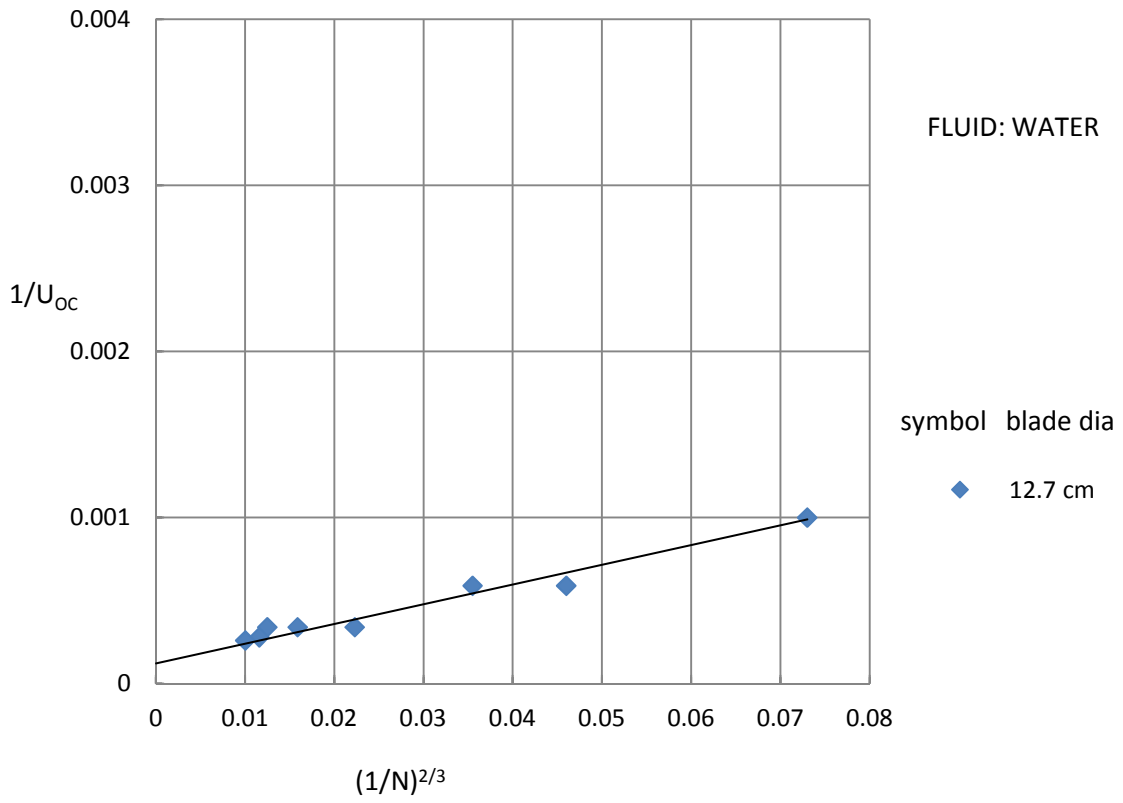
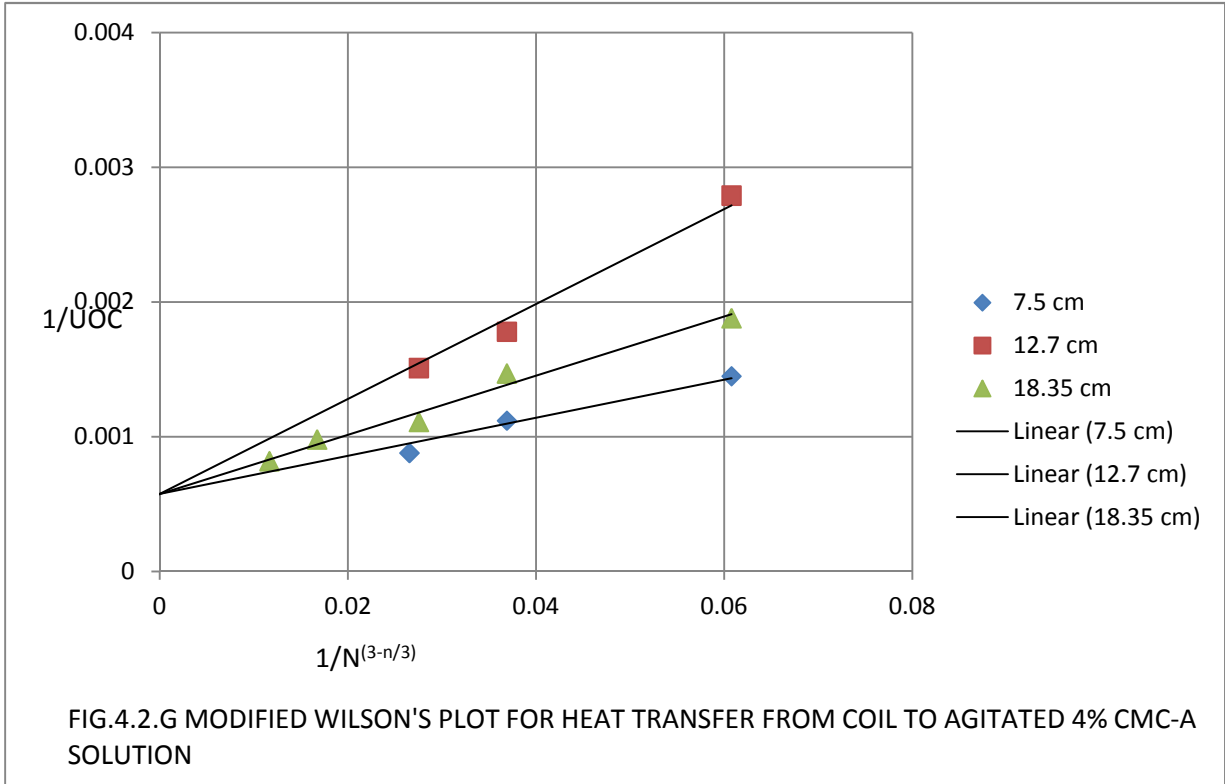
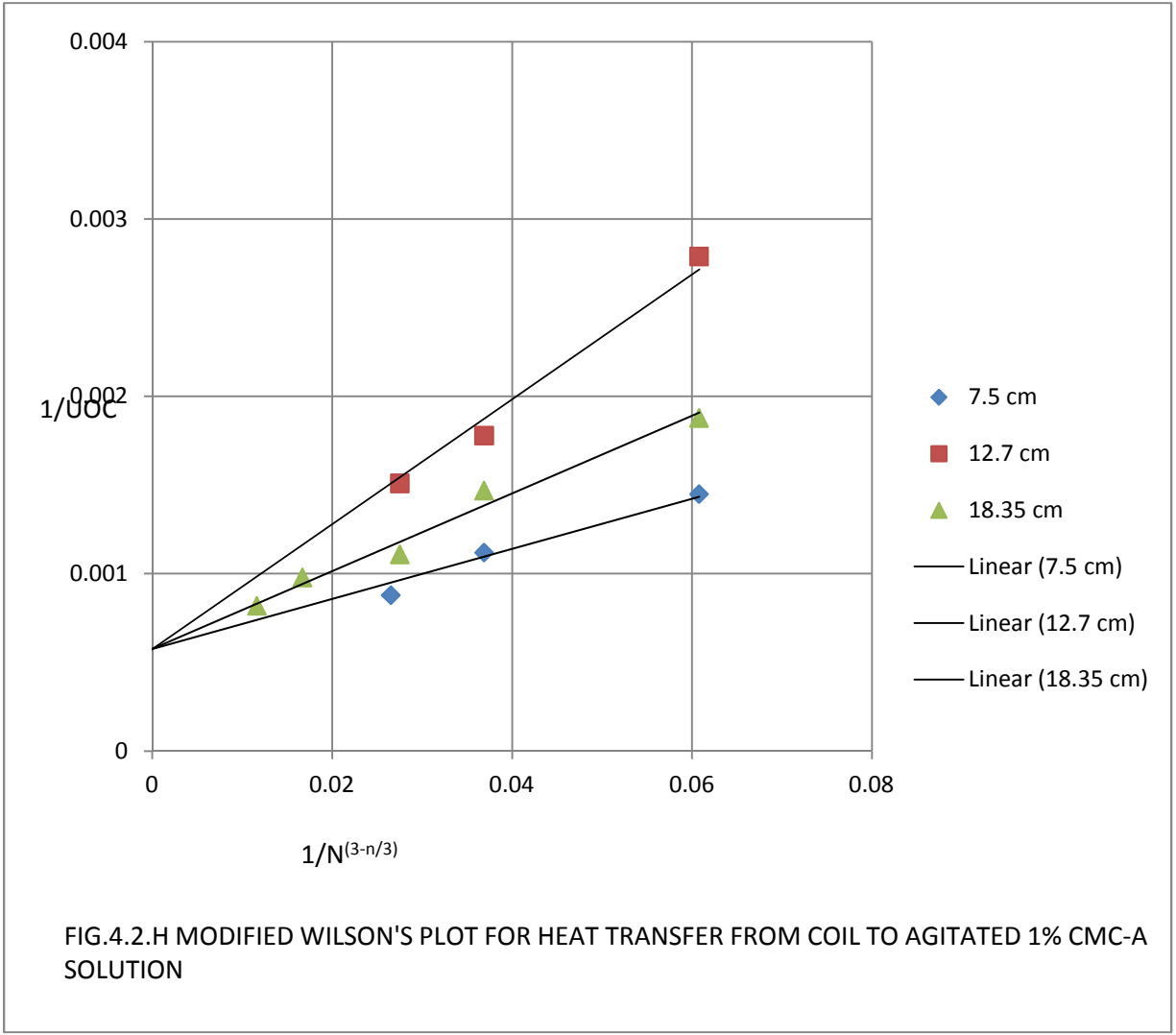
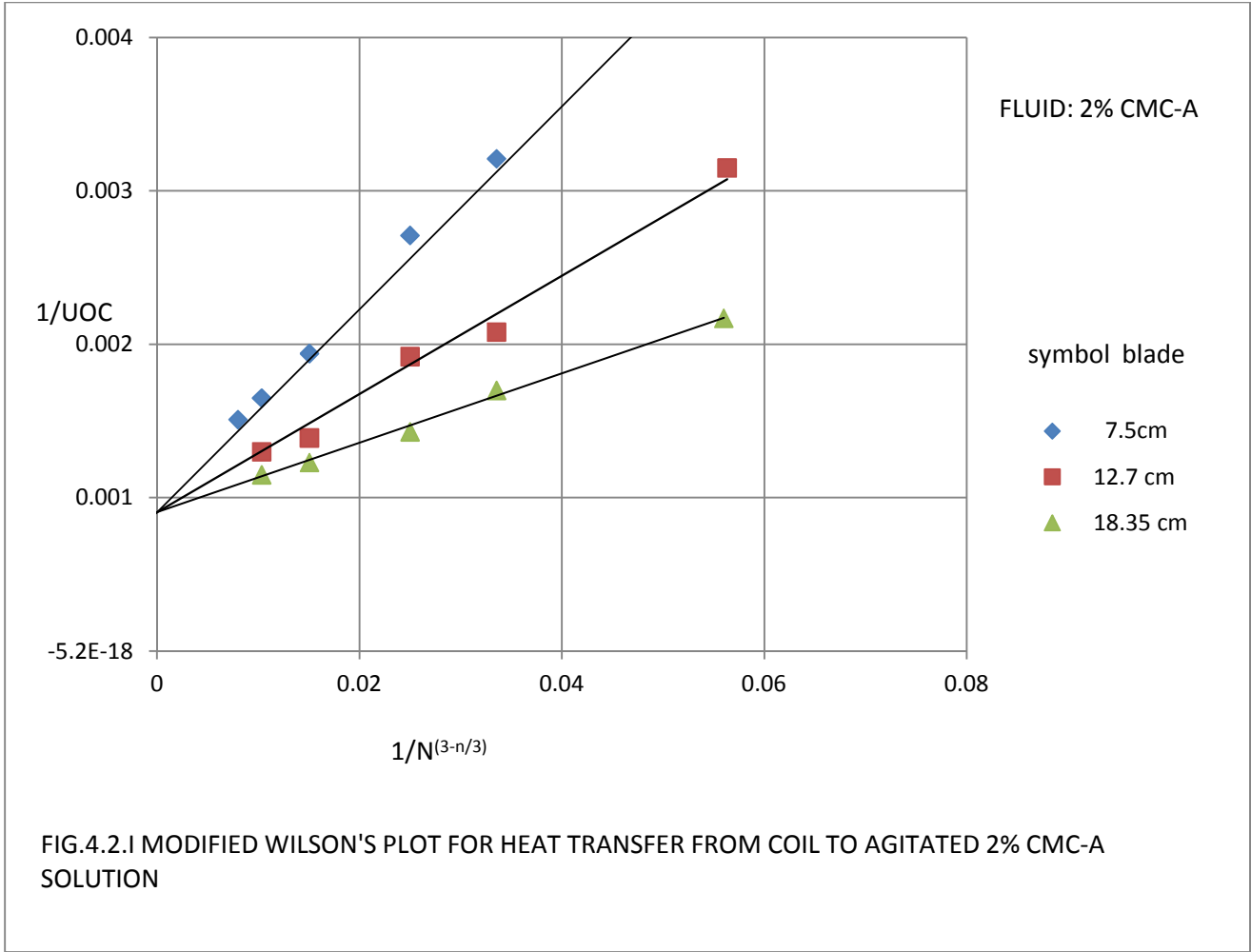


FIG. 4.2.F. MODIFIED WILSON'S PLOT FOR HEAT TRANSFER FROM COIL TO AGITATED WATER







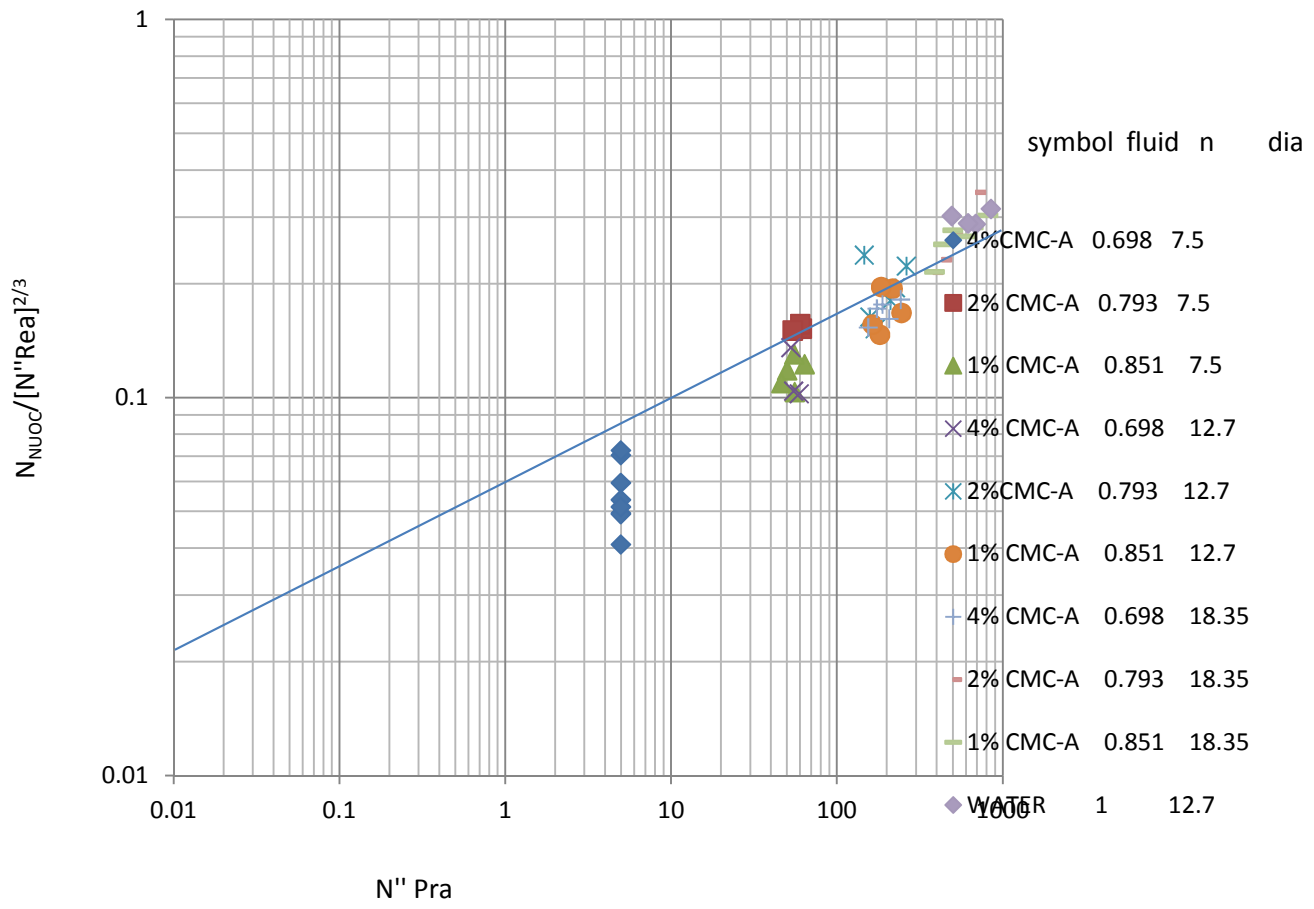


FIG. 4.2.J AGITATED FLUID TO COIL HEAT TRANSFER CORRELATION (EFFECT OF PRANDTL NUMBER)

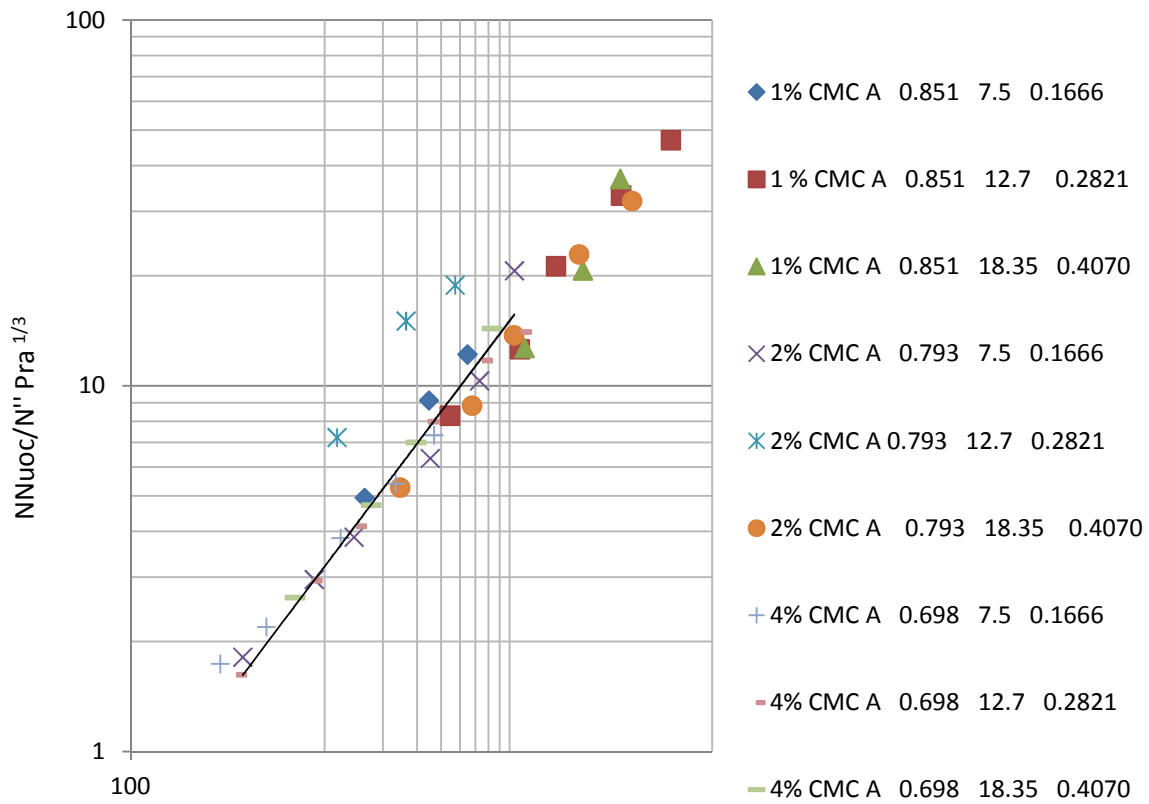


FIG. 4.2.K HEAT TRANSFER CORRELATION FOR COIL TO AGITATED FLUID (EFFECT OF REYNOLDS NUMBER)

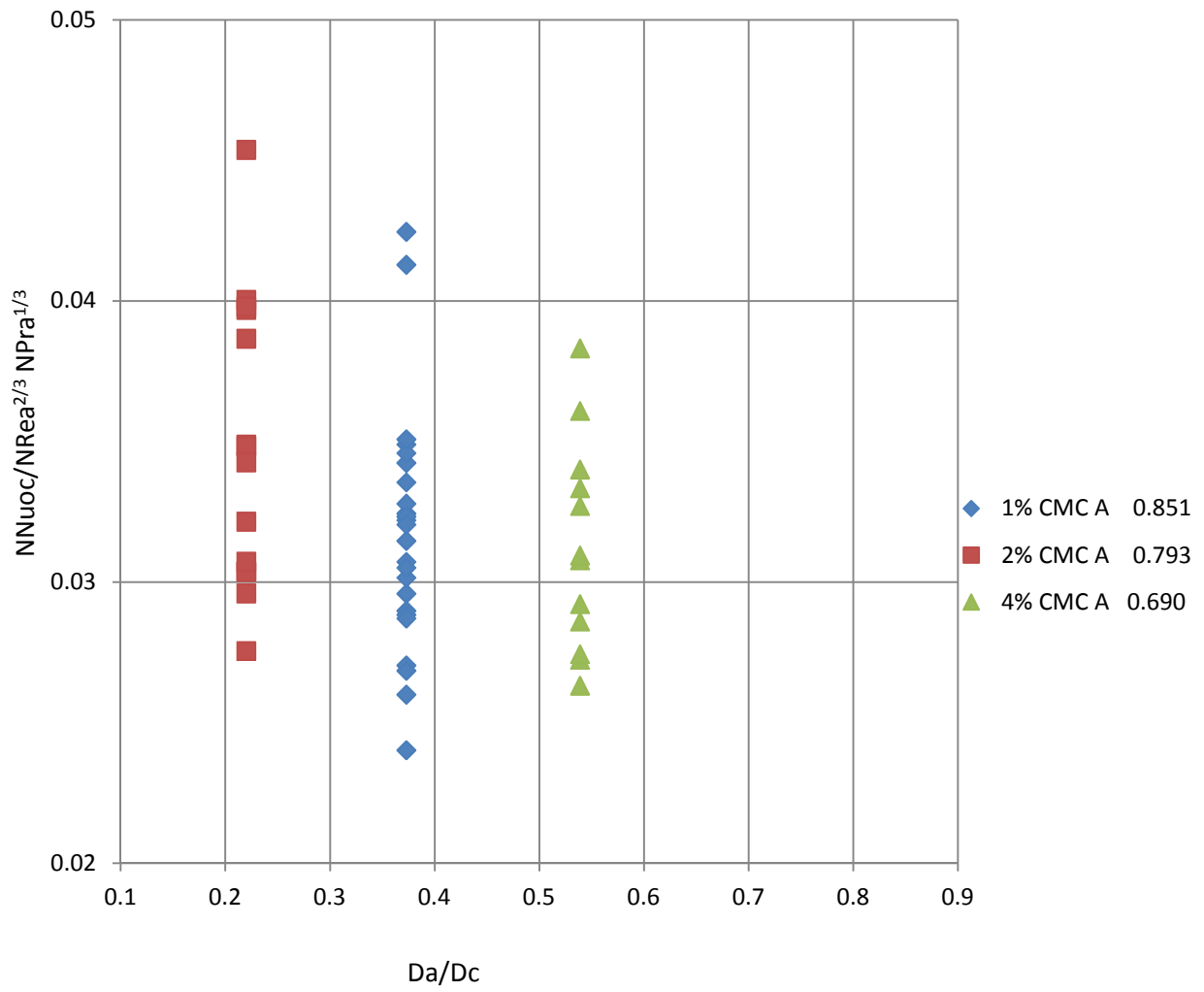


FIG. 4.2.L- HEAT TRANSFER CORRELATION FOR COIL TO AGITATED FLUIDS (EFFECT OF AGITATOR DIAMETER)

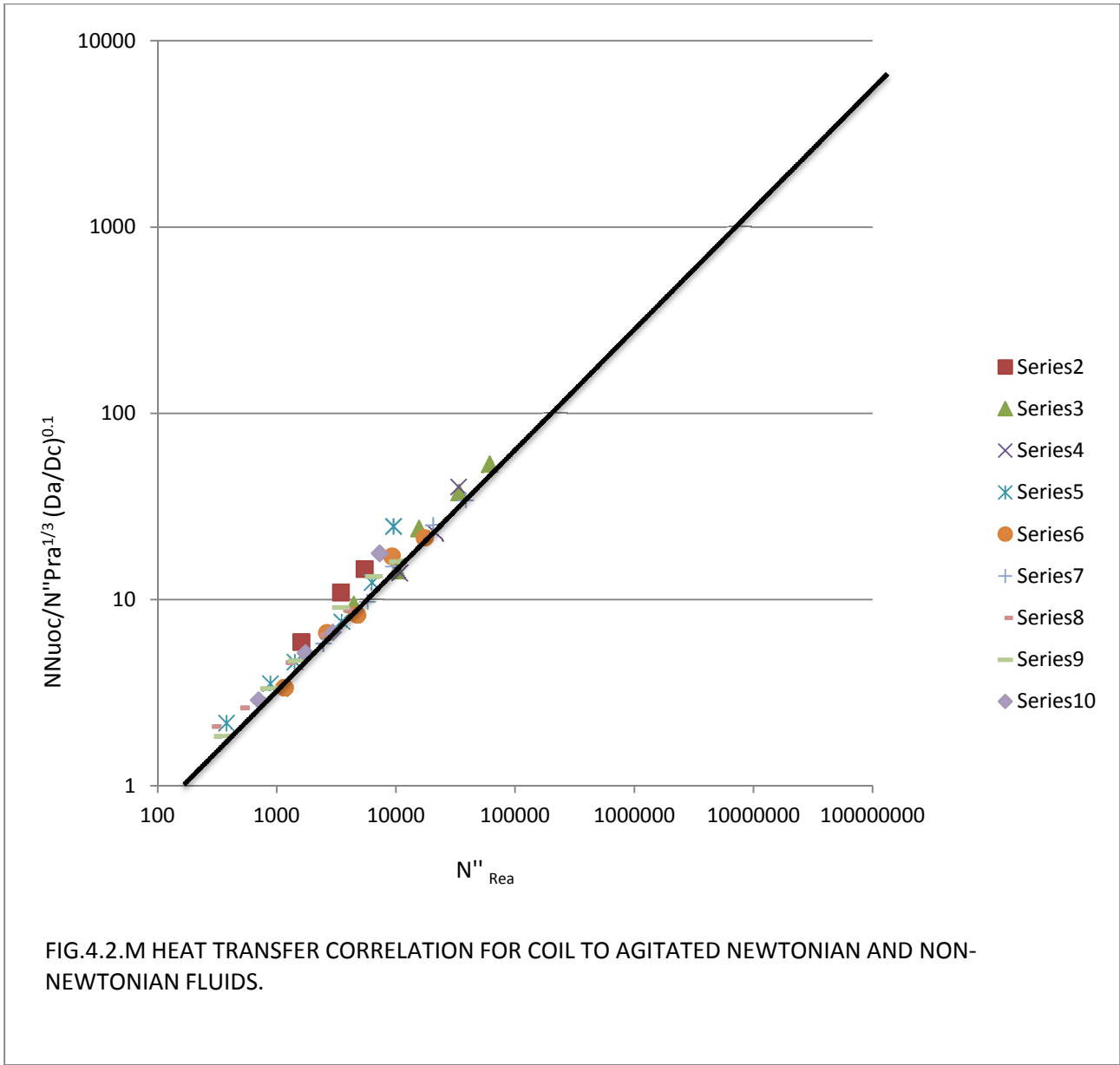


FIG.4.2.M HEAT TRANSFER CORRELATION FOR COIL TO AGITATED NEWTONIAN AND NON-NEWTONIAN FLUIDS.

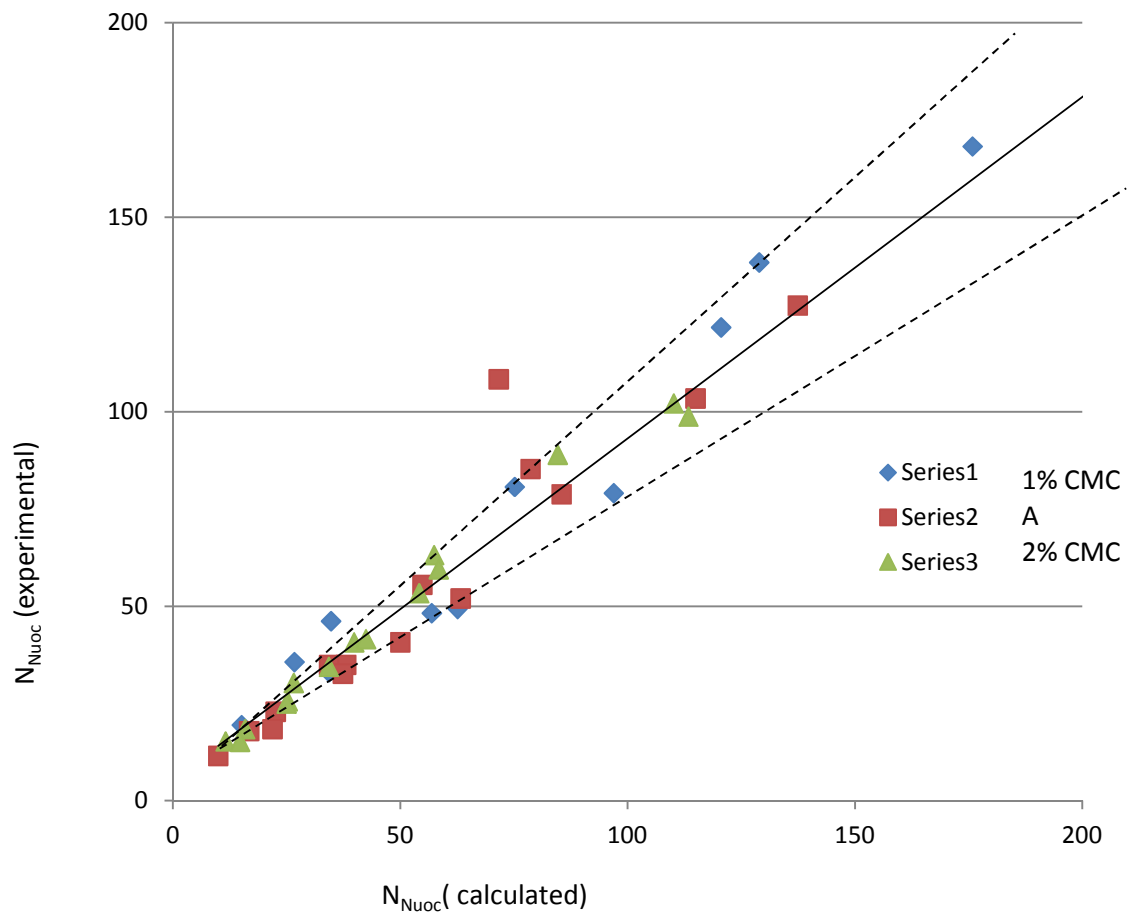


FIG. 4.2.N COMPARISON BETWEEN EXPERIMENTAL & CALCULATED NUSSOLT NUMBER N_{Nuoc} (HEAT TRANSFER FROM COIL TO AGITATED FLUIDS)

$$N_{Nuoc} = C_2 (N''_{Rea})^{2/3} (N''_{Pra})^{1/3} (D_a/D_c)^{0.1} \quad (4.2.3)$$

In order to obtain the constant C_2 of equation (4.2.3) the data of water, and 1,2 and 4% CMC-A solutions for all the three agitators are shown in figure 4.2.M where $N_{Nuoc}/N''_{Pra}{}^{1/3} (D_a/D_c)^{0.1}$ are plotted against N''_{Rea} . The least square line with exponent 2/3 on the Reynolds number, correlating successfully the above data, is drawn and the value of C_2 is found to be 0.036. Thus, equation (4.2.3) can be written as

$$N_{Nuoc} = 0.036 (N''_{Rea})^{2/3} (N''_{Pra})^{1/3} (D_a/D_c)^{0.1} \quad (4.2.4)$$

Experimental values of Nusselt numbers are compared with those calculated from equation (4.2.4) in figure 4.2.N. Equation (4.2.4) correlates 50 data points for $290 < N''_{Rea} < 1.4 \times 10^7$, $4.9 < N''_{Pra} < 850$ and $0.166 < (D_a/D_c) < 0.403$ with a standard deviation of 15.03%.

4.3 Comparison and Discussion

The heat transfer from both, the vessel wall and helical coil submerged in the vessel fluid are mechanically aided heat transfer. Mixing of the heat is caused by the agitation of the fluid. Boundary layer (of course of different thicknesses), formed on the surface, control the resistance. With the coil tubes being near the impeller, one may expect higher coefficient from the coil compared to that from the vessel wall. The mixing of the fluids, a measure of power requirement for agitation, depends not only on Reynolds and Prandtl groups but also on the blade area and its fluid pumping characteristics. Therefore, coefficients will definitely depend on the type and width of the blades. However, if one is interested to compare or to estimate the coefficients of the jacket from the data available for the coil, the ratio (h_j/h_{oc}) can be obtained from equations ((4.2.3) and (4.2.4). for the same Reynolds and Prandtl numbers

$$\frac{h_j}{h_{oc}} = 8.39 (D_c/D_T)^{0.1} (D_t/D_T) \quad (4.3.1)$$

Thus, it is seen that the ratio (h_j/h_{oc}) depends on the diameters of the coil, vessel and coil tube. In the present experimental investigation (h_j/h_{oc}) is found to be 0.405.

The viscosities of the fluids at the coil and vessel surfaces are entirely different. Non-Newtonian behavior of the fluid further complicates the problem as the average or even approximate estimation of stresses at the coil tube outside and vessel surfaces are difficult. The availability of such a procedure for the evaluation of true shear stresses at these surfaces will make the correct

determination of viscosities and their use in Reynolds and Prandtl numbers will make the correlation appropriate and more predictive. Power requirements for the agitation of Newtonian and non-Newtonian fluids as reported by Metzner and Otto (1961) and Calderbank and Moo Young (1961) are correlated in terms of agitator Reynolds number. The deep interrelationship between power requirements and fluid mixing or in turn the heat transfer rate, demonstrates the use of agitator Reynolds number in coefficient correlation. Possibly due to above reasons the agitator Reynolds and corresponding Prandtl numbers have continuously been used by almost all the previous investigators. The Reynolds and Prantal numbers used in the present investigation (discussed in chapter 3) and that used by many earlier investigators (chapter 2) working with power law fluids are compared below:

Reference No.	Reynolds number	Prandtl number
Present work	$\frac{D_a^n (\pi D_a N)^{2-n} \rho}{K n^{1-n} 4^{n-1}}$	$\frac{C_p K}{k} \left(\frac{4\pi N}{n} \right)^{n-1}$
59	$\frac{D_a^2 N^{2-n} \rho}{K(3n + 1/4n)^n 8^{n-1}}$	$\frac{C_p \mu_{d_\infty}}{k}$
66	$\frac{D_a^2 N^{2-n} \rho}{K [1 - \exp(-1.4 \times 10^{-5} D_a^2 N^{2-n} \rho / K)]}$	$\frac{C_p K N^{n-1}}{k} [1 - \exp(-1.4 \times 10^{-5} D_a^2 N^{2-n} \rho / K)]^{-1}$
60	$\frac{D_a^2 N^{2-n} \rho}{K (4\pi)^{n-1}}$	$\frac{C_p K (4\pi N)^{n-1}}{k}$
70	$\frac{D_a^2 N^{2-n} \rho}{K}$	$\frac{C_p K N^{n-1}}{k}$
73	$\frac{D_a^2 N^{2-n} \rho}{K k_s^{n-1}}$	$\frac{C_p K (k_s N)^{n-1}}{k}$
74	$\frac{D_a^2 N^{2-n} B^{1-n} \rho}{K(3n + 1/4n)^n}$	$\frac{C_p K (BN)^{n-1}}{k} \left(\frac{3n + 1}{4n} \right)^n$

Mitsubishi and Miyairi (1973) studied 1.5, 2, 3, 4.5 and 5% CMC solutions and correlated their data using non-Newtonian viscosity of Ellis model at the jacketed vessel wall. By using this modified viscosity, the non-Newtonian data could be correlated in terms of Newtonian heat transfer. The method of Mitsubishi and Miyairi (1973) using viscosity at the surface may not be feasible for a situation in which coil surface is immersed in agitated fluid because the viscosity of non-Newtonian fluids at the vessel surface and coil surface are entirely different. Even approximate estimation of average shear stresses at the coil outside surface and the vessel wall is difficult. The evaluation of true shear stress at these surfaces will result in correct determination of non-Newtonian viscosity and their use in Reynolds and Prandtl groups will be more appropriate. In absence of such a procedure, however, the deep interrelationship between power requirement and fluid mixing or in turn the heat transfer rate either from vessel wall or from the coil, demonstrate the use of agitator Reynolds number in the correlation of heat transfer coefficient for both the vessel wall and coil. Carreau et al. (1966) used the Reynolds number containing viscosity similar to that of pipe flow whereas the Prandtl number used contained the differential viscosity (a constant value) at very high shear rate. They gave separate correlations for Newtonian and non-Newtonian fluids and demonstrated that the use of differential viscosity at high shear rate does not evaluate the proper value of Prandtl number and suggested the evaluation of apparent viscosity from the power curves. Pollard and Kantyka (1969) correlated their data following the procedure of agitator power measurement for non-Newtonian liquids with exponents equal to that used for Newtonian. Their method of determination of heat transfer coefficient needs rheological properties as well as power measurement. In the present approach, however, only rheological properties are required and a single valued correlation for both Newtonian and non-Newtonian fluids can be obtained without much loss of original concept.

Carreau et al. (1966) data were converted in terms of N''_{Rea} and N''_{Pra} . Retaining the exponent of Prandtl number as 1/3 and that of viscosity ratio as 0.14, as $(N_{Nu_j}/N_{Pra}^{1/3})(\mu''_{aw}/\mu''_{ab})^{0.14}$ versus N''_{Rea} . Heating data of non-Newtonian fluids of water are correlated by the following equation with a standard deviation of 16.2%.

$$N_{Nu_j} = 0.495 (N''_{Rea})^{\frac{2}{3}} (N''_{Pra})^{\frac{1}{3}} (\mu''_{aw}/\mu''_{ab})^{0.14} \quad (4.3.2)$$

Cooling data, however, do not agree with the above equation. Further attempt was made to correlate the non-Newtonian data of Sandall and Patel (1970) in terms of N''_{Rea} and N''_{Pra} suggested in the present work. It shows the jacket side heat transfer result of Sandall and Patel.

The logarithmic plot of $N_{Nuj}/N_{Pra}^{n1/3} (\mu''_{aw}/\mu''_{ab})^{0.14}$ against N''_{Rea} resulted in the following correlation:

$$N_{Nuj} = 0.203 (N''_{Rea})^{\frac{2}{3}} (N''_{Pra})^{\frac{1}{3}} (\mu''_{aw}/\mu''_{ab})^{0.14} \quad (4.3.3)$$

Equation (4.3.3) correlated the Newtonian and non-Newtonian data with average deviation of 10.1 and 17.3% respectively. After including the group $(D_a/D_T)^{0.1}$ in equation (4.3.3) one obtains:

$$N_{Nuj} = 0.226 (N''_{Rea})^{\frac{2}{3}} (N''_{Pra})^{\frac{1}{3}} (D_a/D_T)^{0.1} (\mu''_{aw}/\mu''_{ab})^{0.14} \quad (4.3.4)$$

Thus we find that following the present approach one can successfully correlate even the experimental data obtained with other agitated-vessel geometries and the fluids.

4.3.1 Laminar Forced Convective Heat Transfer to Non-Newtonian Fluids in Helical Coil

In order to obtain exponents b_5 and b_6 and constant c_3 in equation (3.5.4), the proposed form of correlation suggested for laminar forced convection to non-Newtonian fluids in helical coils in chapter 3. 0.5, 1 and 2% CMC-A solutions were investigated in a helical coil of 34 cm diameter made from a tube of diameter 1.898 cm. Before conducting the heat transfer runs, pressure drop measurements were made to verify the fitness of the coil. The resulting pressure drop data and calculated variables are presented in table 5.20. Coil flow is compared with capillary shear flow in figure 4.3.A, friction factor, f_c , is shown in figure 4.3.B against the Reynolds number N_{Re_2} defined by equation

$$N_{Re_2} = \frac{D_t U \rho}{\mu_2} \quad (4.3.1.1)$$

Where

$$\mu_2 = \frac{W}{8U'/D_t} = K' \left(8U'/D_t \right)^{n'-1} = K' \left(\frac{\tau_w}{K'} \right)^{(n'-1)/n'} \quad (4.3.1.2)$$

The use of Reynolds number N_{Re_2} seems to be more appropriate since it is defined with a viscosity term evaluated at prevailing wall shear stress. Rheological parameters K' and n' too, are evaluated at τ_w .

For laminar flow of non-Newtonian fluids flowing through straight pipe, all the three definitions of Reynolds number N'_{Re} , N_{Re_1} and N_{Re_2} are found to be the same.

According to present approach the Dean number is defined as

$$N_{D_2} = N_{Re_2} \left(D_t / D_c \right)^{1/2} \quad (4.3.1.3)$$

And for non-Newtonian fluids the correlation of the friction factor will take the following form

$$f_c / f_s = \varphi(N_{D_2}) \quad (4.3.1.4)$$

Where

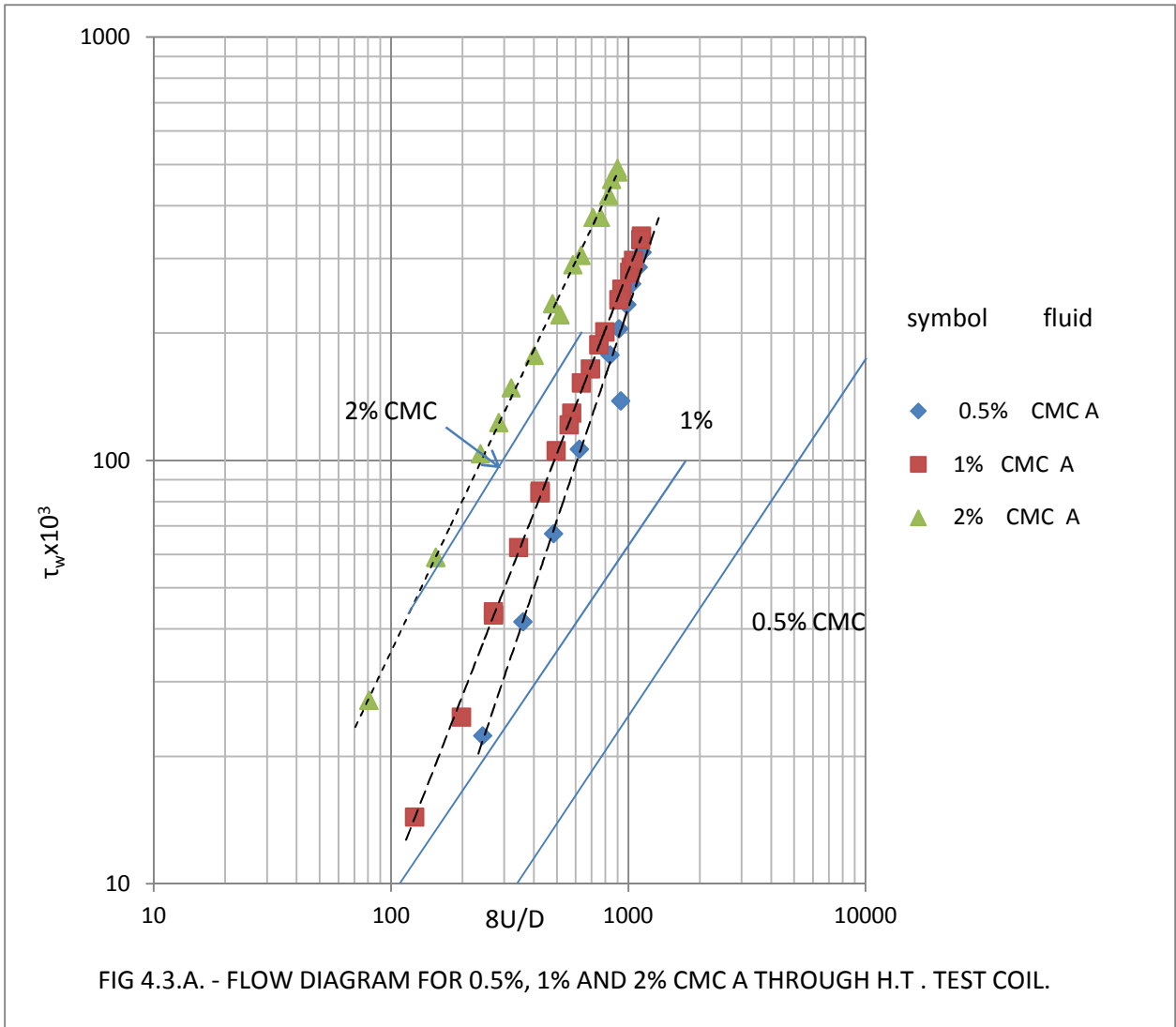
$$f_{SL} = 16 / N_{Re_2}$$

The ratio of friction factor, f_c , for coil to that of straight tube, f_{SL} , evaluated at Reynolds number, N_{Re_2} , from equation

$$f_{SL} = 16 / N_{Re_2}$$

Is plotted against N_{D_2} in figure 4.3.B.

Three non-Newtonian fluids: 0.5 (n= 0.937), 1(n= 0.851) and 2% (n= 0.793) CMC-A in water, representatives of pseudoplastic fluids obeying power law, and water, a Newtonian fluid were investigated. Most of the data obtained were in laminar region and very few data of non-Newtonian fluids could fall in turbulent region. Due to lack of turbulent flow data for non-Newtonian fluids in coil, only laminar region data are processed and correlated.



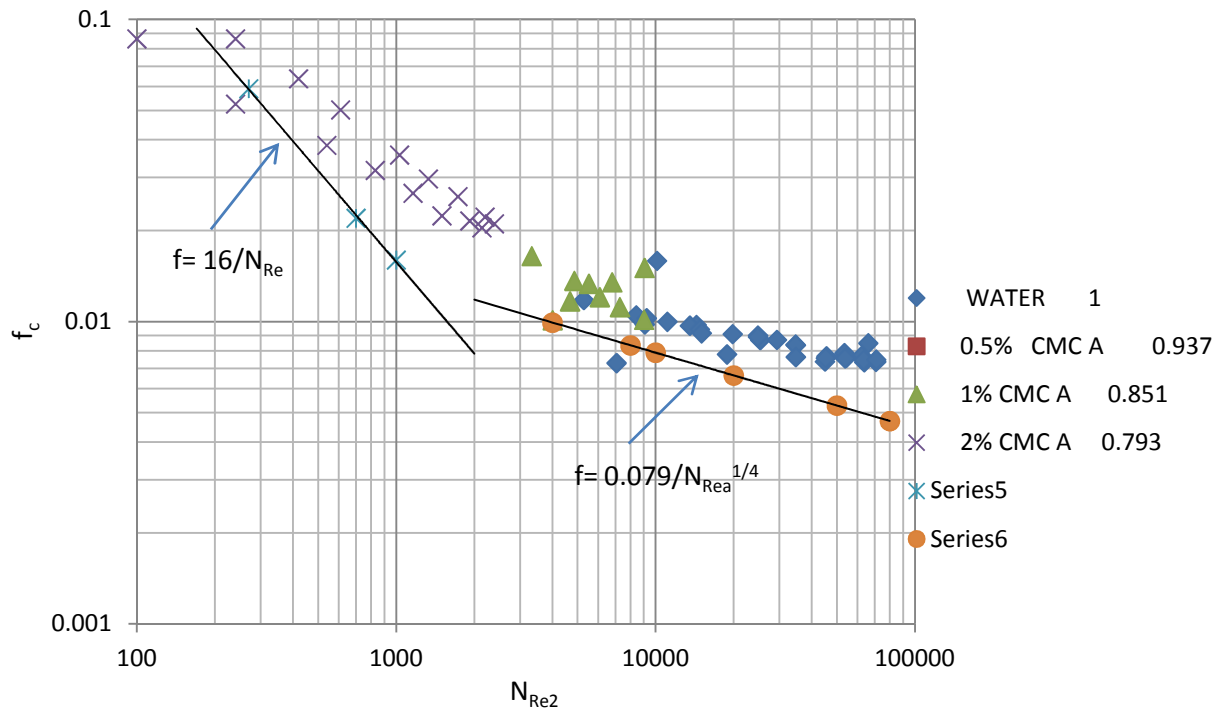
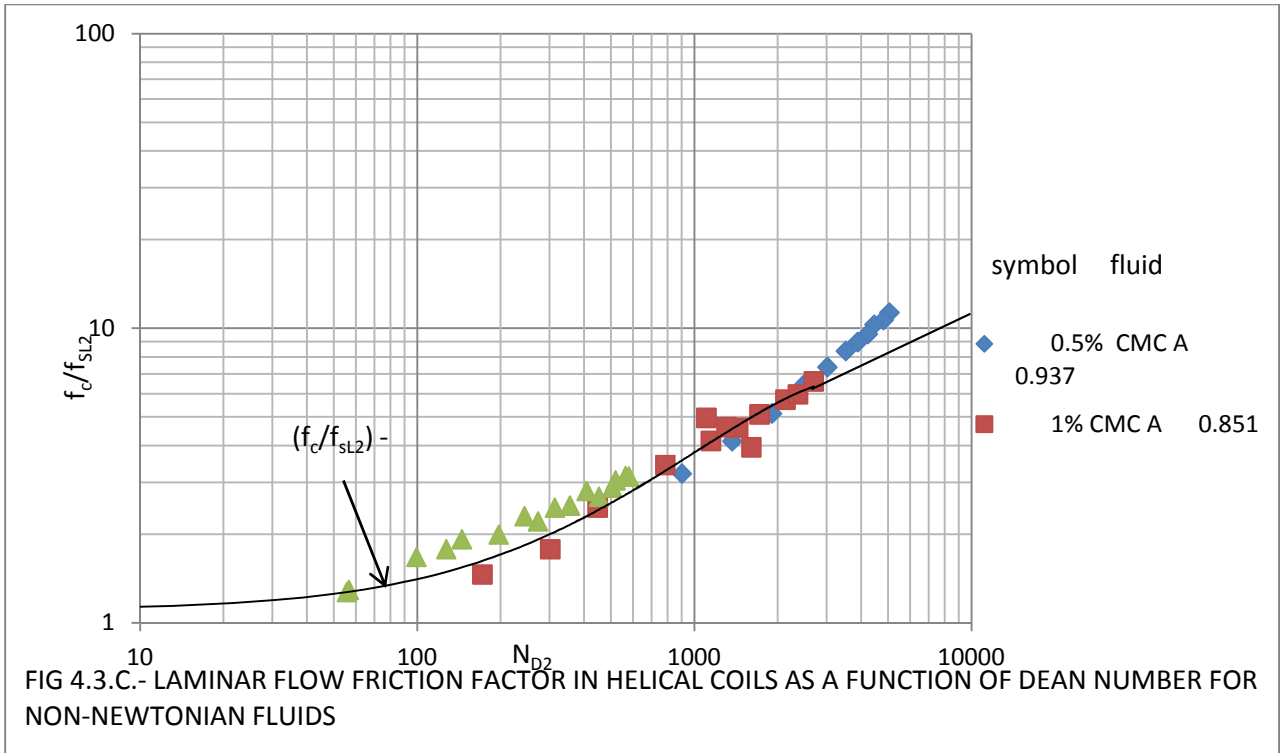


FIG 4.3.B.- VARIATION OF FRICTION FACTOR WITH REYNOLDS NUMBER FOR WATER, 2% CMC A, 1% CMC A & 0.5% CMC A THROUGH H.T. TEST COIL.



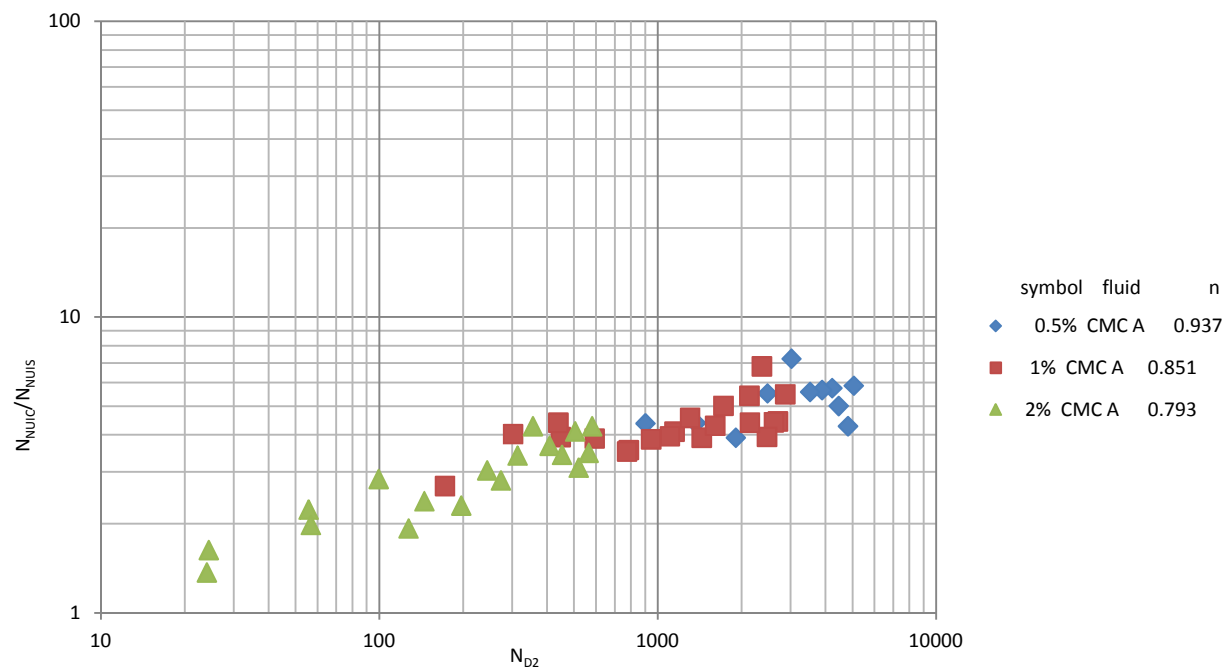
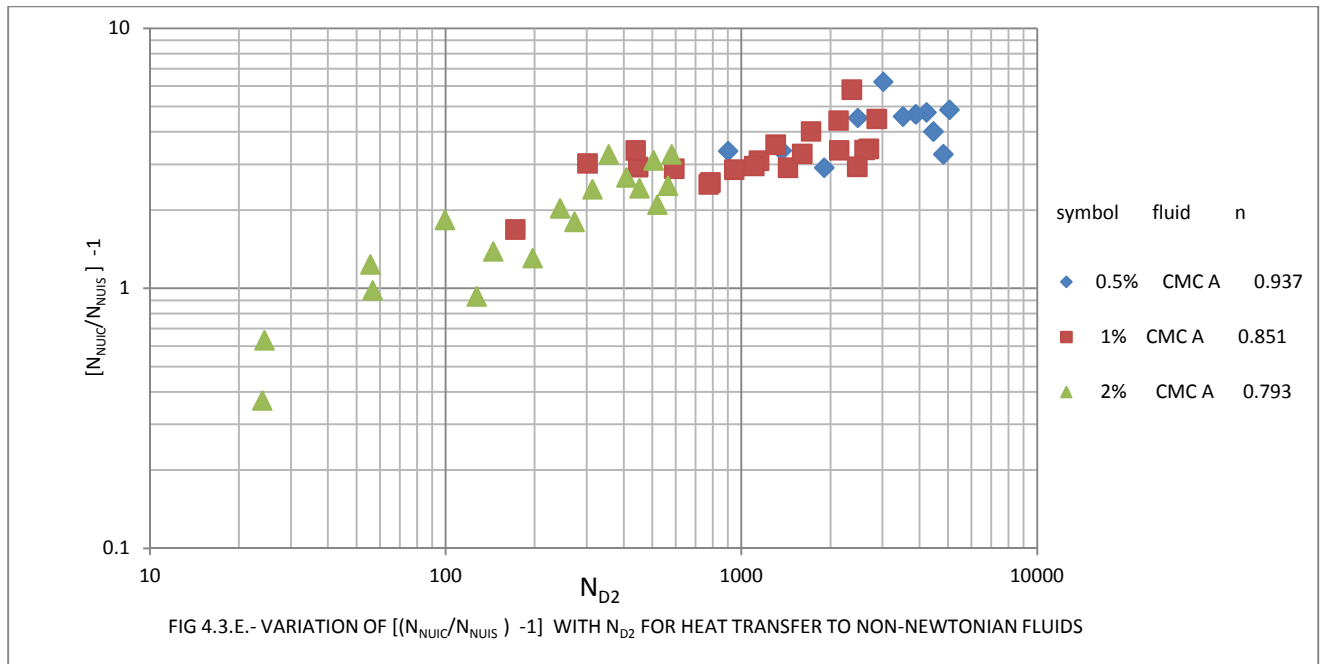


FIG 4.3.D. VARIATION OF N_{NUIC}/N_{NUIS} WITH N_{D2} FOR HEAT TRANSFER TO NON-NEWTONIAN FLUIDS



In chapter 3 it has been shown that the Newtonian and non-Newtonian data over wide range of coil diameter and fluid behavior, can be successfully correlated by the use of apparent viscosity μ_2 (defined by equation (4.3.1.2) evaluated at the wall shear stress in Reynolds number N_{Re_2} . This suggests the possibility of analogues correlation in terms of the ratio (N_{Nuic}/N_{Nuis}) . N_{Nuis} was calculated by equation (3.5.3). Calculated values of N_{Nuic} , N_{Nuis} along with other parameters are presented in table 50 for water, 51 for 0.5% CMC-A, 52 for 1% CMC-A and in 53 for 2% CMC-A.

4.3.2 Effect of Dean Number N_{D2}

In order to find the effect N_{D2} on the ratio of heat transfer in coil to heat transfer in straight pipe, the first attempt was to plot the ratio $(\frac{N_{Nuic}}{N_{Nuis}})$ against N_{D2} on a logarithmic scale as shown in figure (4.3.D).

No direct relationship is seen between the two variables in this manner. At very low Dean Number or very small curvature, effect of secondary flow will be negligible and then N_{Nuic} will be equal to N_{Nuis} . Therefore, the next attempt was to plot $(\frac{N_{Nuic}}{N_{Nuis}}) - 1$ against N_{D2} . This plot is shown in figure 4.3.E

No net conclusion could be drawn even from this figure except for some qualitative ones. The heat transfer coefficient in coil is always higher than in straight pipe as it is seen that $\frac{N_{Nuic}}{N_{Nuis}}$ is always greater than unity. The ratio h_{ic}/h_{is} increases with Dean number, N_{D2} . These data do not

follow any particular trend. This may be explained by the fact that due to non Newtonian behavior Prandtl number changes by change of flow rate affecting the heat transfer rate. Data plotted for three fluids in figure 4.3.E have wide range of Prandtl number. However, an exponent of Dean number $1/2$ correlated the data much better. This trial has been made on the basis of theoretical results presented by previous investigators.

4.3.2 Effect of Prandtl Number

Prandtl number N_{Pr2} was calculated by using apparent viscosity, μ_2 evaluated at wall shear stress, τ_w , obtained from the measured pressure drop.

The μ_2 may be evaluated by the following derived expression

$$\mu_2 = k'(\tau_w/k')^{\frac{n'-1}{n'}}$$

Subsequently $N_{Re2} = DtU\rho/\mu_2$ and

$$\text{Dean Number } N_{D2} = N_{Re2} (D_t / D_c)^{1/2}$$

In order to understand the effect of N_{Pr2} on the ratio h_{ic}/h_{is} , a logarithmic plot of $(\frac{N_{uic}}{N_{uis}})^{-1} N_D^{-1/2}$ versus N_{Pr2} is shown in figure 4.3.F. Figure 4.3.F clearly shows the effect of N_{Pr2} . Many other trials were made by changing the exponent of Dean number N_{D2} , in the ordinate variable of figure 4.3.G.

Values of exponent other than $1/2$ on N_{D2} tend to increase the scatter of data points. The slope of mean line shown represents the exponent of N_{Pr2} and was estimated to be 0.12 for Prandtl number range of 4.9 to 225. Finally $(\frac{N_{uic}}{N_{uis}} - 1) N_{Pr2}^{-0.12}$ is plotted against N_{D2} in figure 4.3.G to verify the exponent of N_{D2} . The mean line passing through data points show an exponent equal to $1/2$. The resulting correlation is obtained as

$$\frac{N_{uic}}{N_{uis}} = 1 + 0.066 N_{D2}^{1/2} N_{Pr2}^{0.12} \quad 4.3.2$$

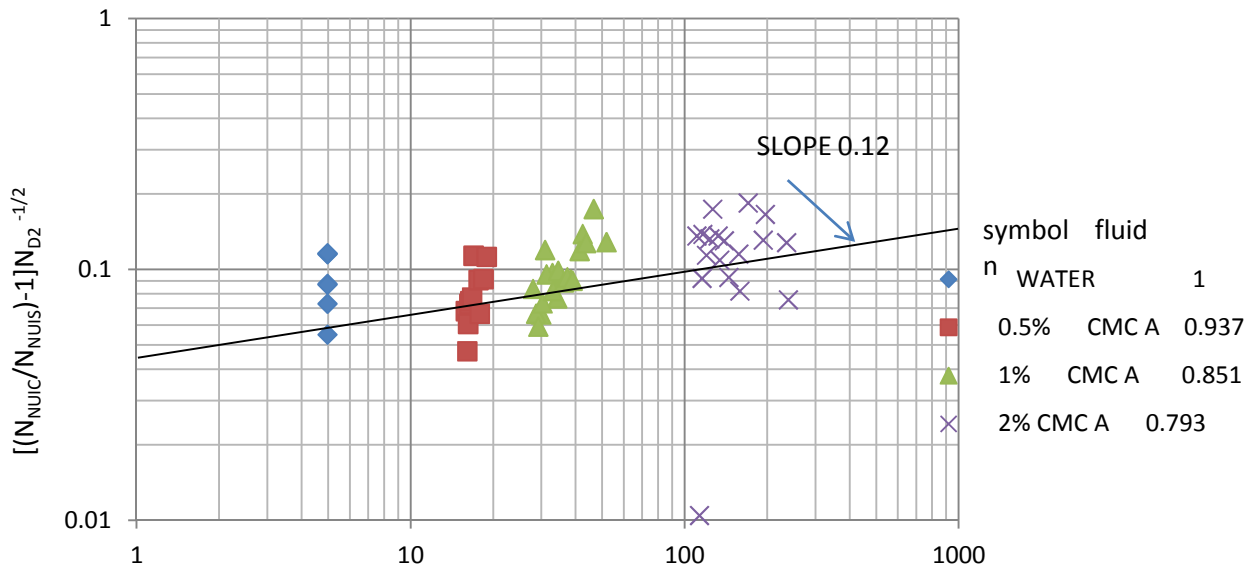


FIG 4.3.F.- LAMINAR FLOW HEAT TRANSFER CORRELATION FOR NON-NEWTONIAN FLUIDS THROUGH COILS : EFFECT OF PRANDTL NUMBER

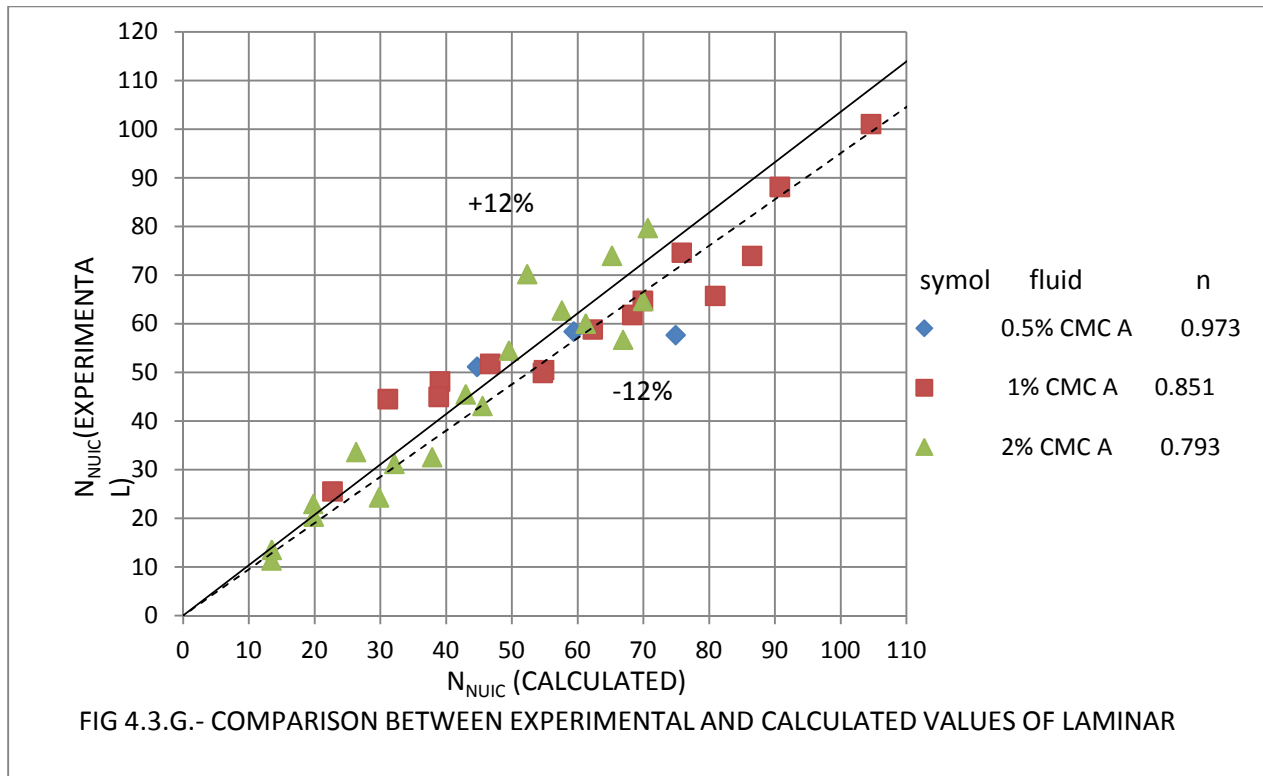


Figure 4.3.G compares the experimental Nusselt number with corresponding values calculated from equation 12. It is found that that laminar flow heat transfer data in the range $24 < N_D < 2000$, $N_{pr2} < 225$ and $0.793 < n \leq 1$ can successfully be correlated using equation 12 with a standard deviation of 15.5%.

According to Ozisik and Topa kaglu's theoretical approach, there is possibly of two regions to exist, namely the heated and cooled region in the fluid space and that too depends upon Prandtl number.

This equation is similar to the equation suggested by Mori and Nakjayama (1965) for flow of Newtonian fluids across tube banks where the ratio $\frac{N_{uic}}{N_{uis}}$ is the function of N_{pr2} and N_{D2} .

CHAPTER 5

SUMMARY AND CONCLUSIONS

As a result of the present investigation on the heat transfer to agitated fluids flowing through immersed helical coils in jacketed vessels, the following conclusions are drawn:

- (1) A modified Wilson plot of $1/U_j$ or $1/U_{oc}$ versus $1/N^{(3-n)/n}$ is appropriate for the determination of individual heat transfer coefficients, h_j and h_{oc} .
- (2) For small temperature driving forces the non isothermal correlation is negligible.
- (3) The heat transfer data for agitated Newtonian and non-Newtonian fluids have been successfully correlated by using the viscosity of the fluid evaluated at the impeller tip assuming a cylinder of diameter equal to that of impeller rotating in an infinite fluid. Data of 1, 2 and 4% CMC-A, for three impeller diameters, have been correlated by the following equations:

for vessel wall

$$N_{Nu_j} = 0.302 N''_{Rea}{}^{2/3} N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_T} \right)^{0.1}$$

(Standard deviation 8.03%)

For coil

$$N_{Nu_{oc}} = 0.036 N''_{Rea}{}^{2/3} N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_c} \right)^{0.1}$$

(Standard deviation 15.03%)

For $290 < N''_{Rea} < 1.4 \times 10^7$, $4.9 < N''_{Pra} < 850$ and $0.166 < \frac{D_a}{D_T} < 0.403$

Using the above concepts of Reynolds and Prandtl numbers it is also possible to correlate the available published data for other non-Newtonian fluids obtained with different impeller geometries.

- (4) The use of same Reynolds and Prandtl number expressions in correlation of jacket side and coil side heat transfer data provides a method of comparison between h_j and h_{oc} . The ratio h_j/h_{oc} is given by

$$\frac{h_j}{h_{oc}} = 8.39 \left(\frac{D_c}{D_T} \right)^{0.1} \left(\frac{D_t}{D_T} \right)$$

- (5) Under the identical flow conditions, laminar flow heat transfer rate in helical coils are higher than those in straight pipes. The ratio N_{Nuic}/N_{Nuis} is found to be a function of N_{D_2} and N_{pr_2} . The data obtained have been correlated by the following equation.

$$N_{Nuic}/N_{Nuis} = 1 + 0.0666 N_{D_2}^{1/2} N_{pr_2}^{0.12}$$

(Standard deviation 15.52%)

$$24 < N_{D_2} < 2000, 40 < N_{pr_2} < 225 \text{ and } 0.793 < n < 1$$

TABLE 5.1.A: LIST OF FLUIDS AND RANGE OF VARIABLES STUDIED FOR HEAT TRANSFER FROM JACKET WALL TO THE AGITATED FLUIDS AND AGITATED FLUIDS TO COIL

Vessel Diameter= 45.0 cm; Coil Tube Diameter = 1.898 cm; Coil Diameter = 34 cm

S. No.	Fluid	n	k	D_a	N	N''_{Rea} X10⁻³	N''_{Pra}	N_{Nuj}	N_{Nuoc}
1	Water	1.0	0.0075	12.70	57-1000	707.4-14140	4.966	383.6-2378.3	470.0-30000.0
2	1% CMC-A	0.851	0.134-0.139	7.50	50-150	1.61-5.46	61.23-54.0	196.3-361.9	19.5-46.2
			0.139-0.143	12.70	50-500	4.43-61.18	63.75-46.2	353.8-1261.2	33.1-168.2
			0.130-0.136	18.35	50-150	10.80-33.57	59.40-52.7	416.2-914.7	49.3-138.4
3	2% CMC-A	0.793	0.940-0.880	7.50	50-700	0.38-9.54	262.00-146.0	88.2-578.0	11.6-108.4
			0.910-0.870	12.70	50-500	1.16-1740	245.00-163.9	172.7-800.4	18.4-103.5
			0.900-0.873	18.35	50-500	2.46-38.60	242.50-154.3	304.9-1771.0	32.7-171.7
4	4% CMC-A	0.698	2.91-2.860	7.50	100-700	0.29-3.67	687.90-381.0	96.2-472.9	15.3-53.4
			3.00-2.780	12.70	50-700	0.35-10.38	812.00-387.0	113.7-876.2	15.2-102.2
			2.91-2.900	18.35	50-300	0.70-7.30	847.00-616.8	193.1-840.8	25.0-113.4

TABLE 5. 1.B: LIST OF FLUIDS AND RANGE OF VARIABLES STUDIED FOR HEAT TRANSFER TO FLUIDS FLOWING THROUGH H.T. TEST COIL.

Coil Tube Diameter = 1.898 cm; Coil Diameter = 34 cm

S. No	Fluid	n	N_{Nuic}	$N_{Re2} \times 10^{-3}$	N_{Pr2}	N_{D2}	N_{Gz}
1	Water	1.0	33.8-1203.96	5.289-70.91	4.966	-	-
2	0.5% CMC-A	0.937	115.42-51.19	3.82-21.40	15.87-18.94	5056.0-902.0	1385.40- 295.15
3	1.0% CMC-A	0.851	25.559-109.1	0.730-12.15	51.778-27.95	172.0-2870.0	154.20- 1382.40
4	2.0% CMC-A	0.793	11.37-79.69	0.1017-2.46	238.90- 110.72	24.0-581.0	99.99- 1120.37

**TABLE 5.2: HEAT TRANSFER FROM JACKET WALL TO VESSEL FLUID AND VESSEL FLUID TO COIL
EXPERIMENTAL DATA**

Jacket Fluid: Water (n=1.0) =22.4 kg/min

Vessel Diameter= 45.25 cm

Coil Fluid: Water (n=1.0)

Coil Tube Diameter = 1.898 cm

Tank Fluid: Water (n=1.0)

Coil Diameter = 34.00 cm

Run No.	R.P.M.	T	T_{jav}	T_{cav}	Q	U_j	U_{oc}	$\frac{1}{U_j}$	$\frac{1}{U_{oc}}$	$\frac{1}{N^{3-n/3}}$	h_j	h_{oc}
1	50	39.30	46.025	33.800	23.10	329.34	997.0	.00303	.00100	.07326	460.83	1136.36
2	100	38.60	46.050	34.050	33.90	436.68	1690.0	.00229	.00059	.04664	699.30	2127.66
3	150	39.10	45.750	34.425	33.90	489.00	1685.6	.00204	.00059	.03546	847.45	2127.66
4	300	39.00	45.900	36.100	36.16	502.46	2931.3	.00199	.00034	.02232	884.96	4545.45
5	500	43.65	44.025	35.200	32.77	673.00	2872.0	.00148	.00035	.01587	1573.00	4400.00
6	700	43.50	45.050	36.125	36.16	827.00	2940.0	.00120	.00034	.01269	2900.00	4550.00
7	800	42.00	47.075	39.075	44.07	832.60	3490.0	.00120	.00028	.01160	2941.18	5985.00
8	1000	41.40	46.100	38.975	46.50	826.19	3880.0	.00121	.00026	.01000	2857.14	7250.00

**TABLE 5.3: HEAT TRANSFER FROM JACKET WALL TO VESSEL FLUID AND VESSEL FLUID TO COIL
EXPERIMENTAL DATA**

Jacket Fluid: Water =22.4 kg/min

Vessel Diameter= 45.25 cm

Coil Fluid: 1% CMC-A (n=0.851)

Coil Tube Diameter = 1.898 cm

Tank Fluid: 1% CMC-A (n=0.851)

Coil Diameter = 34.00 cm

Run No.	R.P.M	T	T_{jav}	T_{cav}	Q	U_j	U_{oc}	$\frac{1}{U_j}$	$\frac{1}{U_{oc}}$	$\frac{1}{N^{3-n/3}}$	h_j	h_{oc}
Agitator Diameter = 7.5 cm												
1	50	53.40	66.050	38.025	23.45	117.50	358.0	0.00564	0.00279	0.06079	235.85	451.46
2	100	51.65	64.730	38.640	31.12	228.00	361.0	0.00438	0.00178	0.03690	335.57	826.41
3	150	51.20	63.535	39.515	34.74	270.00	658.7	0.00370	0.00151	0.02754	434.78	1069.50
Agitator Diameter = 12.7 cm												
1	50	51.20	62.700	39.950	25.58	213.26	531.0	0.00468	0.00188	0.06079	304.88	766.28
2	100	52.60	63.925	40.575	35.17	297.70	679.0	0.00335	0.00147	0.03690	512.80	1117.30
3	150	51.00	62.125	40.875	39.50	341.00	902.0	0.00293	0.00111	0.02754	653.59	1869.10
4	300	50.10	59.600	41.025	42.90	432.97	1094.0	0.00230	0.00098	0.01672	1111.10	2941.10
5	300	50.60	59.700	41.775	46.00	485.00	1220.0	0.00206	0.00082	0.01165	1515.15	4065.00
Agitator Diameter = 18.35 cm												
1	50	54.55	66.500	42.215	36.62	293.80	688.0	0.00340	0.00145	0.06079	500.00	1142.85
2	100	53.50	65.225	41.575	45.96	375.80	892.0	0.00266	0.00112	0.03690	793.65	1831.50
3	150	52.40	63.710	41.960	50.87	431.20	1126.0	0.00231	0.00088	0.02754	1098.90	3205.10

**TABLE 5.4: HEAT TRANSFER FROM JACKET WALL TO VESSEL FLUID AND VESSEL FLUID TO COIL
EXPERIMENTAL DATA**

Jacket Fluid: Water =21.318 kg/min

Vessel Diameter= 45.25 cm

Coil Fluid: 2% CMC-A (n=0.793)

Coil Tube Diameter = 1.898 cm

Tank Fluid: 2% CMC-A (n=0.793)

Coil Diameter = 34.00 cm

Run No.	R.P. M.	T	T_{jav}	T_{cav}	Q	U_j	U_{oc}	$\frac{1}{U_j}$	$\frac{1}{U_{oc}}$	$\frac{1}{N^{3-n/3}}$	h_j	h_{oc}
Agitator Diameter = 7.5 cm												
1	50	55.00	70.450	40.060	14.380	89.20	233.0	0.01122	.00448	.05633	105.930	280.11
2	100	55.20	69.425	40.635	19.494	131.40	311.0	0.00761	.00321	.03355	171.526	433.84
3	150	55.40	68.650	41.225	22.550	163.50	368.0	0.00610	.00271	.02500	231.480	554.00
4	300	56.90	69.300	42.790	29.312	226.60	515.0	0.00475	.00194	.01506	336.700	966.18
5	500	55.60	67.580	42.500	34.420	275.40	606.0	0.00363	.00165	.01034	546.540	1342.28
6	700	56.02	67.535	42.800	37.205	310.00	652.0	0.00322	.00151	.00800	694.440	1652.90
Agitator Diameter = 12.7 cm												
1	50	56.95	70.500	41.370	21.318	150.80	317.0	0.00660	.00315	.05633	207.400	445.53
2	100	56.50	70.300	41.750	30.665	213.00	380.0	0.00469	.00208	.03355	343.640	843.88
3	150	59.25	70.400	43.185	29.455	275.00	521.0	0.00363	.00192	.02500	458.710	985.22
4	300	54.20	64.900	42.765	35.370	317.00	715.0	0.00315	.00139	.01506	729.920	1061.85
5	500	54.75	64.850	43.515	37.335	354.00	768.5	0.00282	.00130	.01034	961.400	2500.00
Agitator Diameter = 18.35 cm												
1	50	57.40	71.000	41.570	31.500	222.50	460.7	0.00451	.00217	.05633	366.300	790.51
2	100	57.80	70.335	42.775	38.025	291.50	585.9	0.00344	.00170	.03355	602.410	1257.80
3	150	58.00	69.950	43.750	42.750	329.30	694.5	0.00303	.00143	.02500	813.000	1904.76
4	300	56.30	67.360	43.590	44.567	386.35	814.0	0.00258	.00123	.01506	1250.000	3076.90
5	500	56.50	68.500	43.375	55.600	445.00	872.0	0.00225	.00115	.01034	2127.600	41493.37

**TABLE 5. 5: HEAT TRANSFER FROM JACKET WALL TO VESSEL FLUID AND VESSEL FLUID TO COIL
EXPERIMENTAL DATA**

Jacket Fluid: Water =21.318 kg/min

Vessel Diameter= 45.25 cm

Coil Fluid: 0.5% CMC-A (n=0.937)

Coil Tube Diameter = 1.898 cm

Tank Fluid: 4% CMC-A (n=0.698)

Coil Diameter = 34.00 cm

Ru n No.	R.P. M.	T	T_{jav}	T_{cav}	Q	U_j	U_{oc}	$\frac{1}{U_j}$	$\frac{1}{U_{oc}}$	$\frac{1}{N^{3-n/3}}$	h_j	h_{oc}
Agitator Diameter = 7.5 cm												
1	100	50.92	62.750	39.870	15.268	93.50	318.0	0.01070	0.00315	.04201	115.6 0	369.68
2	150	51.80	64.975	40.880	17.500	127.0 0	372.2	0.00787	0.00269	.03194	171.8 2	445.43
3	300	50.57	64.420	39.340	26.880	186.0 8	553.0	0.00537	0.00181	.01976	301.2 5	732.60
4	500	50.60	64.140	40.075	31.080	220.1 0	683.0	0.00454	0.00146	.01404	401.6 0	985.22
5	700	50.98	64.050	40.925	35.700	261.8 0	820.0	0.00381	0.00122	.01111	568.1 8	1290.32
Agitator Diameter = 12.7 cm												
1	50	52.80	67.400	40.860	16.200	106.6 0	315.5	0.00937	0.00317	.06779	136.6 1	366.97
2	100	52.50	66.975	40.860	24.300	168.5 0	483.2	0.00593	0.00206	.04201	257.7 3	619.19
3	150	52.60	66.330	41.550	29.150	203.0 0	608.0	0.00492	0.00164	.03194	348.4 3	833.33
4	300	50.69	63.250	41.635	35.700	272.5 0	902.7	0.00366	0.00110	.01976	621.1 2	1526.70
5	500	49.59	61.525	41.795	38.850	312.0 0	1087. 8	0.00320	0.00091	.01404	869.5 6	2150.50
6	700	50.53	62.325	42.585	40.950	332.8 0	1163. 4	0.00310	0.00085	.01111	1052. 63	2469.13
Agitator Diameter = 18.35 cm												
1	50	51.10	65.825	39.480	24.200	157.2 0	472.1	0.00636	0.00210	.06779	232.0 2	604.22
2	100	51.00	65.265	39.560	34.300	229.0 0	695.0	0.00436	0.00144	.04201	432.9 0	1005.00
3	150	50.55	64.110	40.125	39.204	277.0 0	870.5	0.00361	0.00114	.03194	641.0 3	1438.80
4	300	51.10	62.750	43.325	39.900	328.2 7	1228. 2	0.00304	0.00081	.01976	1010. 10	2739.70

TABLE 5.6: HEAT TRANSFER FROM JACKET TO THE AGITATED FLUID CALCULATED VARIABLES

Jacket Fluid: Water; Vessel Fluid: Water; $D_a = 12.7$ cm

Run No.	R.P. M.	N_{Nuj}	K	μ'_a	$N''_{Rea} X 1$	N''_{Pra}	$\frac{N_{Nuj}}{N''_{Rea}{}^{2/3}}$	$\frac{N_{Nuj}}{N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuj}}{N''_{Rea}{}^{2/3} N''_{Pra}}$	$\frac{N_{Nuj}}{N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_T}\right)^{0.1}}$
1	50	383.6	0.0075	-	7.07	4.966	0.4855	224.85	0.28462	255.33
4.966	4.966	4.966	4.966	4.966	4.966	4.966	0.4656	341.23	0.27296	387.46
4.966	4.966	4.966	4.966	4.966	4.966	4.966	0.4301	413.49	0.25213	469.55
4.966	4.966	4.966	4.966	4.966	4.966	4.966	0.2833	431.79	0.26062	490.33
4.966	4.966	4.966	4.966	4.966	4.966	4.966	0.3559	767.87	0.20808	871.98
4.966	4.966	4.966	4.966	4.966	4.966	4.966	0.5245	1414.42	0.30748	1606.10
4.966	4.966	4.966	4.966	4.966	4.966	4.966	0.5835	1435.03	0.28474	1629.57
4.966	4.966	4.966	4.966	4.966	4.966	4.966	0.4900	1674.74	0.23972	1901.77

TABLE 5.7: HEAT TRANSFER FROM JACKET TO THE AGITATED FLUID CALCULATED VARIABLES

Jacket Fluid: Water; Vessel Fluid: 1% CMC-A

Run No.	R.P.M.	N_{Nuj}	K	μ'_a	$N''_{Rea} \times 10^3$	N''_{Pra}	$\frac{N_{Nuj}}{N''_{Rea}{}^{2/3}}$	$\frac{N_{Nuj}}{N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuj}}{N''_{Rea}{}^{2/3} N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuj}}{N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_T}\right)^{0.1}}$
Agitator Diameter = 7.5 cm										
1	50	196.32	0.1340	.0924	1.610	61.23	1.5330	49.83	0.38927	59.12
2	100	274.27	0.1384	.0860	3.458	59.95	1.2248	71.42	0.30765	85.48
3	150	361.91	0.1395	.0816	5.466	54.07	1.1788	95.74	0.31187	114.60
Agitator Diameter = 12.7 cm										
1	50	253.78	0.1395	.0962	4.430	63.75	0.9399	63.60	0.23557	72.18
2	100	426.86	0.1350	.0839	10.170	55.56	0.9160	111.45	0.23916	126.49
3	150	544.05	0.1400	.0819	15.630	55.24	0.8931	143.17	0.23081	162.49
4	300	924.89	0.1430	.0754	33.880	49.96	0.8979	252.01	0.24467	286.03
5	500	1261.23	0.1425	.0697	61.180	46.18	0.8190	352.30	0.22876	400.97
Agitator Diameter = 18.35 cm										
1	50	416.21	0.1300	.0897	10.770	59.41	0.8634	106.99	0.22197	117.04
2	100	660.64	0.1330	.0826	21.540	54.74	0.8692	172.94	0.22755	189.19
3	150	914.73	0.1360	.0796	33.570	52.71	0.8967	243.78	0.22851	266.69

TABLE 5.8: HEAT TRANSFER FROM JACKET TO THE AGITATED FLUID CALCULATED VARIABLES

Jacket Fluid: Water; Vessel Fluid: 2% CMC-A

Ru n No	R.P. M.	N_{Nuj}	K	μ'_a	$N''_{Rea} \times 10^{-5}$	N''_{Pra}	$\frac{N_{Nuj}}{N''_{Rea}{}^{2/3}}$	$\frac{N_{Nuj}}{N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuj}}{N''_{Rea}{}^{2/3} N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuj}}{N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_T}\right)^0}$
Agitator Diameter = 7.5 cm										
1	50	88.17	0.675	0.3956	0.379	262.0	1.6955	13.798	0.26534	16.516
2	100	142.78	0.670	0.3397	0.855	224.9	1.5519	23.520	0.25567	28.153
3	150	192.68	0.668	0.3173	1.420	210.0	1.5170	32.437	0.25541	38.820
4	300	280.27	0.635	0.2566	3.515	169.9	1.2185	50.773	0.22075	60.776
5	500	449.95	0.655	0.2384	6.306	157.8	1.3230	83.633	0.24598	100.105
6	700	578.06	0.650	0.2205	9.546	146.0	1.2706	110.106	0.24199	131.790
Agitator Diameter = 12.7 cm										
1	50	172.69	0.632	0.3704	1.160	245.0	1.5698	27.670	0.25158	31.404
2	100	286.05	0.645	0.3270	2.635	216.5	1.5892	47.830	0.26574	54.281
3	150	381.83	0.588	0.2740	4.720	181.4	1.3731	67.703	0.24350	76.839
4	300	607.59	0.690	0.2788	9.278	184.6	1.3967	107.158	0.24634	121.624
5	500	800.39	0.680	0.2475	17.402	163.9	1.2127	146.590	0.22211	166.381
Agitator Diameter = 18.35 cm										
1	50	304.90	0.625	0.3663	2.456	242.5	1.6938	49.019	0.27232	53.623
2	100	501.45	0.615	0.3118	5.771	206.4	1.5570	85.135	0.26439	93.148
3	150	676.75	0.610	0.2843	9.500	188.0	1.5139	118.106	0.26422	129.202
4	300	1040.50	0.650	0.2626	20.560	173.9	1.4060	187.140	0.25289	204.705
5	500	1771.03	0.640	0.2330	38.610	154.3	1.5810	330.404	0.29501	361.403

TABLE 5.9: HEAT TRANSFER FROM JACKET TO THE AGITATED FLUID CALCULATED VARIABLES

Jacket Fluid: Water; Vessel Fluid: 4% CMC-A

Ru n No	R.P. M.	N_{Nuj}	K	μ''_a	$N''_{Rea} \times 10^{-5}$	N''_{Pra}	$\frac{N_{Nuj}}{N''_{Rea}{}^{2/3}}$	$\frac{N_{Nuj}}{N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuj}}{N''_{Rea}{}^{2/3} N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuj}}{N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_T}\right)^{0.1}}$
Agitator Diameter = 7.5 cm										
1	100	96.23 2	2.9 1	1.038 8	0.29	687. 9	2.197 0	10.935	0.24966	13.08
2	150	143.0 2	2.8 6	0.903 0	0.51	598. 0	2.270 0	17.026	0.27025	20.38
3	300	250.7 2	2.9 4	0.754 2	1.20	499. 0	2.238 5	31.736	0.28336	37.98
4	500	334.3 0	2.9 4	0.648 7	2.33	429. 0	1.890 3	44.336	0.25190	53.11
5	700	472.9 8	2.9 1	0.578 8	3.67	381. 0	1.895 7	65.419	0.26220	78.30
Agitator Diameter = 12.7 cm										
1	50	113.7 0	2.7 8	1.226 8	0.35	812. 0	2.283 0	12.226	0.24549	13.90
2	100	214.5 3	2.8 1	1.003 1	0.86	664. 0	2.236 2	24.573	0.27063	27.88
3	150	290.0 3	2.8 1	0.887 2	1.46	587. 0	2.248 2	34.775	0.26957	39.46
4	300	517.0 3	2.9 4	0.754 2	3.45	499. 0	2.267 6	65.446	0.28704	74.27
5	500	723.8 0	3.0 0	0.661 9	6.55	438. 0	2.068 0	95.488	0.27282	108.38
6	700	876.2 2	2.9 5	0.584 7	10.38	387. 0	1.844 6	120.52 5	0.25373	136.80
Agitator Diameter = 18.35 cm										
1	50	193.1 3	2.9 0	1.279 8	0.70	847. 0	2.438 5	20.393	0.25749	22.31
2	100	360.3 5	2.9 1	1.038 8	1.74	687. 0	2.493 7	40.948	0.28338	44.79
3	150	533.6 1	2.9 5	0.931 4	2.96	616. 8	2.590 2	62.776	0.30474	68.67
4	300	810.8 2	2.9 0	0.743 9	7.31	492. 0	2.242 0	106.56 7	0.28418	116.58

TABLE 5.10: COMPARISON BETWEEN EXPERIMENTAL AND CALCULATED NESSELT NUMBERS FOR HEAT TRANSFER FROM JACKET WALL TO AGITATED FLUIDS

Run No.	Fluid	Da	R.P.M.(N)	N _{Nuj} (Exprimental)	N _{Nuj} (Calculated)
1	2	3	4	5	6
1	Water	12.7	50	383.6	360.18
			100	582.1	569.9
			150	705.43	747.7
			300	1156.0	1185.4
			500	1310.0	1677.8
			700	2413.0	2097.3
			800	2448.27	2297.9
			1000	2378.0	2658.0
2	1% CMC-A	7.5	50	196.32	127.23
			100	274.27	224.9
			150	361.916	293.0
		12.7	50	253.78	287.6
			100	426.86	476.5
			150	544.05	629.3
			300	924.89	1009.0
			500	1261.23	1472.0
		18.35	50	416.2	517.0
			100	660.64	800.7
			150	914.73	1058.0
3	2% CMC-A	7.5	50	88.17	83.83
			100	142.78	140.89
			150	192.68	190.3
			300	280.27	320.3
			500	449.95	461.5
			700	578.06	602.6
		12.7	50	172.69	183.26
			100	286.05	287.39
			150	381.83	418.6
			300	607.59	658.5
			500	800.39	962.0
		18.35	50	304.9	308.7
			100	501.45	523.0
			150	676.75	706.4
			300	1040.5	1135.0
4	4% CMC-A	7.5	500	1771.0	1656.0
			100	96.23	97.25
			150	143.02	133.52
			300	250.72	223.25
			500	334.3	334.8
		12.7	700	472.98	455.15
			50	113.7	123.7
			100	214.53	211.8

			150	290.03	287.5
			300	517.03	481.4
			500	723.8	709.06
			700	867.22	922.95
		18.35	50	190.13	206.85
			100	360.35	350.7
			150	533.6	482.9
			300	840.82	786.4

TABLE 5.11: HEAT TRANSFER FROM AGITATED FLUID TO THE CALCULATED VARIABLES

Coil Fluid: Water

Vessel Fluid: Water

Run No.	R.P.M. (N)	N_{Nuoc}	μ''_a	$N''_{Rea} \times 10$	N''_{Pra}	$\frac{N_{Nuoc}}{N''_{Rea}{}^{2/3}}$	$\frac{N_{Nuoc}}{N''_{Pra}{}^{1/2}}$	$\frac{N_{Nuoc}}{N''_{Rea}{}^{2/3} N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuoc}}{N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_c}\right)^{0.1}}$
Agitator Diameter = 12.7 cm									
1	50	470.0	0.0075	7.07	4.966	0.05953	275.69	0.03490	321.130
2	100	880.6	0.0075	14.00	4.966	0.07045	516.19	0.04129	601.275
3	150	880.6	0.0075	21.20	4.966	0.05369	516.19	0.03147	601.275
4	300	1883.6	0.0075	42.40	4.966	0.07251	1106.00	0.04246	1288.293
5	500	1820.4	0.0075	70.70	4.966	0.04950	1066.00	0.02898	1241.701
6	700	1885.3	0.0075	99.00	4.966	0.04090	1105.00	0.02402	1287.103
7	800	2482.6	0.0075	113.60	4.966	0.04920	1454.05	0.02884	1693.651
8	100	3000.8	0.0075	141.40	4.966	0.05140	1757.00	0.03016	2046.592

TABLE 5.12: HEAT TRANSFER FROM AGITATED FLUID TO THE COIL CALCULATED VARIABLES

Coil Fluid: 1% CMC-A (n=0.851)

Vessel Fluid: 1% CMC-A (n= 0.851)

Run No.	R.P. M.	N_{Nuoc}	K	μ_a''	$N''_{Rea} \times 1$	N''_{Pra}	$\frac{N_{Nuoc}}{N''_{Rea}{}^{2/3}}$	$\frac{N_{Nuoc}}{N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuoc}}{N''_{Rea}{}^{2/3} N''_{Pra}}$	$\frac{N_{Nuoc}}{N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_c}\right)^{0.1}}$
Agitator Diameter = 7.5 cm										
1	50	19.498	0.1340	0.09246	1.610	61.23	0.1523	4.948	0.03866	5.922
2	100	35.693	0.1384	0.08600	3.458	59.95	0.1565	9.128	0.04004	10.926
3	150	46.190	0.1395	0.08165	5.466	54.07	0.1505	12.196	0.03980	14.598
Agitator Diameter = 12.7 cm										
4	50	33.095	0.1395	0.09626	4.430	63.75	0.1226	8.294	0.03072	9.413
5	100	48.256	0.1350	0.08390	10.170	55.56	0.1035	12.598	0.02704	14.298
6	150	80.726	0.1400	0.08190	15.630	55.24	0.1302	21.244	0.03424	24.110
7	300	121.730	0.1430	0.07544	33.880	49.96	0.1182	33.169	0.03220	37.640
8	500	168.250	0.1425	0.06973	61.180	46.18	0.1093	46.997	0.03051	53.330
Agitator Diameter = 18.35 cm										
9	50	49.359	0.1301	0.08970	10.770	59.41	0.1024	12.698	0.02632	13.880
10	100	79.100	0.1330	0.08266	21.540	54.74	0.1041	20.707	0.02724	22.650
11	150	138.428	0.1360	0.07960	33.570	52.71	0.1357	36.814	0.03609	40.270

TABLE 5.13: HEAT TRANSFER FROM AGITATED FLUID TO THE COIL CALCULATED VARIABLES

Coil Fluid: 2% CMC-A (n=0.793)

Vessel Fluid: 2% CMC-A (n= 0.793)

Run No	R.P. M.	N_{Nuoc}	K	μ'_a	$N''_{Rea} \times 10^{-3}$	N''_{Pra}	$\frac{N_{Nuoc}}{N''_{Rea}^{2/3}}$	$\frac{N_{Nuoc}}{N''_{Pra}^{1/3}}$	$\frac{N_{Nuoc}}{N''_{Rea}^{2/3} N''_{Pra}^{1/3}}$	$\frac{N_{Nuoc}}{N''_{Pra}^{1/3} \left(\frac{D_a}{D_c}\right)^{0.1}}$
Agitator Diameter = 7.5 cm										
1	50	11.593	0.675	0.3956	0.379	2.620	0.2229	1.814	0.03489	2.171
2	100	17.956	0.670	0.3397	0.885	2.250	0.1952	2.958	0.03215	3.540
3	150	22.930	0.668	0.3173	1.420	2.100	0.1806	3.860	0.03039	4.620
4	300	34.990	0.635	0.2566	3.515	1.699	0.1521	6.339	0.02755	7.586
5	500	55.550	0.655	0.2384	6.306	1.578	0.1634	10.325	0.03036	12.350
6	700	108.410	0.650	0.2205	9.546	1.460	0.2383	20.650	0.04538	24.718
Agitator Diameter = 12.7 cm										
7	50	18.436	0.632	0.3704	1.160	2.450	0.1676	2.941	0.02685	3.354
8	100	34.928	0.645	0.3270	2.635	2.165	0.1942	5.841	0.03244	6.629
9	150	40.778	0.588	0.2740	4.720	1.814	0.1467	7.230	0.02600	8.260
10	300	85.339	0.690	0.2788	9.278	1.846	0.1962	15.051	0.03459	17.080
11	500	103.470	0.680	0.2475	17.400	1.639	0.1561	18.864	0.02871	21.406
Agitator Diameter = 18.35 cm										
12	50	32.719	0.625	0.3663	2.456	2.425	0.1818	5.277	0.02922	5.773
13	100	52.060	0.615	0.3118	5.770	2.064	0.1617	8.838	0.02744	9.668
14	150	78.837	0.610	0.2843	9.500	1.880	0.1764	13.759	0.03078	15.050
15	300	127.350	0.650	0.2626	20.560	1.739	0.1721	22.905	0.03095	25.051
16	500	171.740	0.640	0.2330	38.600	1.543	0.1533	32.040	0.02860	34.053

TABLE 5.14: HEAT TRANSFER FROM AGITATED FLUID TO THE COIL CALCULATED VARIABLES

Coil Fluid: 0.5% CMC-A (n=0.973)

Vessel Fluid: 4% CMC-A (n= 0.698)

Ru n No	R.P. M.	N_{Nuoc}	K	μ_a''	$N''_{Rea} \times 10^{-}$	$N''_{Pra} \times 10^{-}$	$\frac{N_{Nuoc}}{N''_{Rea}{}^{2/3}}$	$\frac{N_{Nuoc}}{N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuoc}}{N''_{Rea}{}^{2/3} N''_{Pra}{}^{1/3}}$	$\frac{N_{Nuoc}}{N''_{Pra}{}^{1/3} \left(\frac{D_a}{D_c}\right)^0}$
Agitator Diameter = 7.5 cm										
1	100	15.30 0	2.9 1	1.03 88	0.290	6.88	0.34 93	1.738 6	0.03969	2.083
2	150	18.43 6	2.8 6	0.90 30	0.501	5.98	0.29 26	2.194 8	0.03483	2.626
3	300	30.32 0	2.9 4	0.75 42	1.210	4.99	0.27 07	3.838 0	0.03426	4.594
4	500	40.77 8	2.9 4	0.64 87	2.331	4.29	0.23 17	5.408 2	0.03073	6.473
5	700	53.40 6	2.9 1	0.57 68	3.672	3.81	0.21 47	7.336 0	0.02960	8.781
Agitator Diameter = 12.7 cm										
6	50	15.11 8	2.7 8	1.22 68	0.353	8.12	0.30 36	1.625 6	0.03279	1.844
7	100	25.62 8	2.8 1	1.00 31	0.864	6.64	0.28 22	2.935 6	0.03233	3.331
8	150	34.49 0	2.8 1	0.88 72	1.466	5.87	0.26 74	4.135 5	0.03205	4.693
9	300	63.19 0	2.9 4	0.75 42	3.451	4.99	0.27 71	7.998 7	0.03508	9.077
10	500	89.00 9	3.0 0	0.66 19	6.550	4.38	0.25 43	11.74 10	0.03355	13.324
11	700	102.1 97	2.9 5	0.58 47	10.381	3.87	0.21 52	14.05 71	0.02959	15.951
Agitator Diameter = 18.35 cm										
12	50	20.00 8	2.9 0	1.27 98	0.705	8.47	0.31 58	2.640 8	0.03334	2.889
13	100	41.59 6	2.9 1	1.03 88	1.741	6.87	0.28 79	4.726 8	0.03271	5.170
14	150	59.55 0	2.9 5	0.93 14	2.960	6.17	0.28 91	7.005 9	0.03401	6.664
15	300	113.3 97	2.9 0	0.74 39	7.304	4.92	0.30 24	14.37 24	0.03832	17.723

TABLE 5.15: COMPARISON BETWEEN EXPERIMENTAL AND CALCULATED NESSELT NUMBERS FOR HEAT TRANSFER FROM AGITATED FLUIDS TO COIL

Run No.	Fluid	D _a	R.P.M.(N)	N _{Nuj} (Experimental)	N _{Nuj} (Calculated)
1	2	3	4	5	6
1	Water	12.7	50	470.339	430.8
			100	880.63	682.0
			150	880.63	894.4
			300	1883.6	1417.9
			500	1820.0	2006.9
			700	1885.0	2508.7
			800	2480.0	2748.6
			1000	3000.0	3179.5
2	1% CMC-A	7.5	50	19.498	15.12
			100	35.693	26.74
			150	46.19	34.8
		12.7	50	33.095	34.36
			100	48.256	56.93
			150	80.726	75.19
			300	121.73	120.58
			500	168.25	175.87
		18.35	50	49.359	62.62
			100	79.1	96.96
			150	138.428	128.96
3	2% CMC-A	7.5	50	11.593	9.968
			100	17.956	16.75
			150	22.93	22.63
			300	34.99	38.088
			500	55.55	54.87
			700	108.41	71.66
		12.7	50	18.436	21.896
			100	34.928	34.33
			150	40.778	50.01
			300	85.339	78.67
			500	103.47	114.94
		18.35	50	32.719	37.39
			100	52.06	63.3
			150	78.838	85.5
			300	127.35	137.42
			500	171.47	200.5
4	4% CMC-A	7.5	100	15.3	11.579
			150	18.436	15.898
			300	30.32	26.58
			500	40.778	39.867
			700	53.406	54.19
		12.7	50	15.188	14.77
			100	25.628	25.28
			150	34.49	34.3

			300	63.19	57.45
			500	89.009	84.63
			700	102.197	110.158
		18.35	50	25.008	25.008
			100	41.596	42.47
			150	59.55	58.48
			300	98.82	113.396

TABLE 5.16: HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL EXPERIMENTAL DATA

Jacket Fluid: Water

Coil Fluid: Water

Vessel Fluid: Water

Run No.	U	T	T _{cav}	Q	U _{oc}	h _{ic}	K	μ ₂	τ _w X10 ⁴	$\frac{8U}{D_t}$
1	2	3	4	5	6	7	8	9	10	11
Agitator Diameter = 12.7 cm; R.P.M.= 100										
1	27.96	41.5	35.60	24.86	894.61	1868.8	-	0.0075	29.286	117.87
2	33.26	41.4	35.10	22.60	788.90	1511.2	-	0.0075	59.323	140.20
3	43.86	40.5	34.82	24.86	987.50	2218.3	-	0.0075	98.371	184.87
4	56.81	40.0	34.70	29.38	1222.30	3544.8	-	0.0075	160.698	239.45
5	74.47	39.5	34.65	31.64	1498.78	6426.7	-	0.0075	221.523	313.90
6	100.08	39.2	34.30	31.64	1445.69	5527.9	-	0.0075	446.050	421.80
7	137.17	39.0	34.05	31.64	1416.59	5282.6	-	0.0075	802.740	578.19
8	177.79	38.6	34.05	33.90	1690.00	10362.7	-	0.0075	1189.920	749.38
9	211.35	38.5	34.05	31.64	1680.80	10362.7	-	0.0075	1793.030	896.86
10	246.21	38.2	34.00	31.60	1888.00	26666.6	-	0.0075	2379.840	1037.76
11	261.10	38.0	33.96	31.64	1752.00	12562.8	-	0.0075	2950.350	1100.50

Contd...(Table 5.16)

Run No.	U	T	T _{cav}	Q	U _{oc}	h _{ic}	K	μ ₂	τ _w X10 ⁴	$\frac{8U}{D_t}$
1	2	3	4	5	6	7	8	9	10	11
Agitator Diameter = 12.7 cm; R.P.M.= 300										
1	20.90	45.9	37.85	22.45	688.36	968.24	-	.0075	26.28	88.09
2	36.50	44.2	37.5	29.76	1028.2	1592.6	-	.0075	69.78	153.80
3	53.57	43.1	37.7	32.76	1404.4	2447.4	-	.0075	141.90	225.79
4	58.58	42.0	37.45	32.83	1670.3	3253.09	-	.0075	165.20	246.91
5	79.77	41.0	37.1	32.52	1930.3	4168.4	-	.0075	221.52	336.23
6	98.02	40.2	36.7	33.3	2202.5	5282.6	-	.0075	441.49	413.17
7	136.88	39.5	36.45	34.875	2647.0 6	8210.18	-	.0075	802.70	576.90
8	180.15	39.0	36.1	36.72	2931.2 6	10362.7	-	.0075	1221.39	759.33
9	214.89	38.5	35.97	34.68	3179.5 0	14044.9 0	-	.0075	1793.03	905.75
10	251.98	38.2	35.92	36.38	3701.9 0	26666.6 0	-	.0075	2379.79	1062.10
11	279.65	38.0	36.65	33.25	3275.4 7	15923.5 1	-	.0075	2934.05	1178.70

Contd...(Table 5.16)

Run No.	U	T	T _{cav}	Q	U _{oc}	h _{ic}	K	μ ₂	τ _w g _c	$\frac{8U}{D_t}$
1	2	3	4	5	6	7	8	9	10	11
Agitator Diameter = 18.35 cm										
1	35.913	46.0	37.475	32.635	886.21	1811.59	-	0.0075	64.58	151.37
2	40.03	45.10	37.20	31.28	916.60	1898.60	-	0.0075	129.90	168.72
3	59.46	43.70	37.47	35.86	1333.58	4321.50	-	0.0075	165.20	250.62
4	78.30	43.50	37.25	38.57	1428.60	5282.60	-	0.0075	281.59	330.03
5	98.025	43.00	36.97	39.12	1503.38	6097.50	-	0.0075	436.29	413.27
6	115.98	42.70	36.65	41.37	1583.00	7209.80	-	0.0075	596.99	488.858
7	137.176	42.50	37.40	41.94	1591.60	7680.50	-	0.0075	733.50	578.19
8	178.977	42.00	36.07	44.08	3490.00	10362.70	-	0.0075	1238.80	754.38
9	212.535	41.40	35.90	46.90	1975.30	34482.70	-	0.0075	1744.13	895.83
10	250.803	41.10	36.12	44.73	2081.40	82644.60	-	0.0075	2396.14	1057.12
11	280.240	41.00	36.25	40.46	1971.80	34482.70	-	0.0075	2999.30	1181.20

TABLE 5.17: HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL EXPERIMENTAL DATA

Jacket Fluid: Water

Coil Fluid: 0.5% CMC-A (n= 0.937)

Vessel Fluid: 4% CMC-A (n= 0.698)

Run No.	U	T	T _{cav}	Q	U _{oc}	h _{ic}	K	μ ₂	τ _w g _c	$\frac{8U}{D_t}$
1	2	3	4	5	6	7	8	9	10	11
Agitator Diameter = 18.35 cm										
1	57.70	51.55	40.675	37.40	657.1	1466.3	0.0435	0.02860	22.40	243.20
2	85.38	50.82	40.300	37.40	701.9	1673.1	0.0440	0.02780	41.61	359.87
3	114.82	50.32	39.675	39.06	699.7	1652.9	0.0442	0.02699	67.21	483.96
4	147.50	50.10	39.100	39.06	843.5	2529.7	0.0450	0.02669	106.42	621.70
5	173.11 7	49.77	39.890	37.81	950.6	3501.4	0.0440	0.02561	138.43	929.68
6	198.73	49.72	39.705	42.18	881.5	2832.8	0.0440	0.02528	177.63	837.60
7	216.69	49.60	40.100	40.31	896.7	2974.4	0.0440	0.02493	204.84	913.34
8	233.76	49.70	39.965	43.42	905.8	3058.1	0.0440	0.02473	233.65	985.28
9	244.95	49.81	39.905	39.27	856.3	2702.7	0.0440	0.02455	261.64	1032.40
10	261.44	50.10	40.265	42.59	825.6	2377.5	0.0439	0.02429	286.45	1101.90
11	270.86	50.15	40.530	41.97	929.8	3350.8	0.0435	0.03966	310.46	1141.66

TABLE 5.18: HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL EXPERIMENTAL DATA

Jacket Fluid: Water

Coil Fluid: 1% CMC-A (n= 0.851)

Tank Fluid: 1% CMC-A (n=0.851)

Ru n No.	U	T	T_{cav}	Q	U_{oc}	h_{ic}	K	μ₂	τ_wg_c	$\frac{8U}{D_t}$
1	2	3	4	5	6	7	8	9	10	11
Agitator Diameter = 12.7 cm										
1	29.84	54.2	41.400	29.84	407.4	732.0	0.170	0.07820	14.39	125.77
2	46.96	54.2	42.050	31.97	567.8	1275.5	0.168	0.07022	24.78	197.93
3	64.13	53.7	41.600	31.97	589.3	1379.3	0.169	0.06405	43.17	270.30
4	100.8 5	53.1	40.875	32.61	606.0	1446.7	0.1715	0.05796	84.75	425.07
5	137.0 0	52.6	40.575	35.17	679.0	1853.9	0.1720	0.05400	129.50	577.44
6	164.4 0	52.2	40.75	35.17	684.2	1883.2	0.1718	0.05171	164.70	692.93
7	188.8 9	52.1	40.875	38.37	768.4	2525.2	0.1715	0.04270	201.47	796.16
8	122.7 0	51.8	41.375	38.37	805.8	2895.2	0.1700	0.04726	254.20	938.67
9	243.6 9	51.6	41.025	40.50	869.3	3710.5	0.1710	0.04668	286.22	1027.10
10	569.3 4	50.9	41.150	34.10	767.8	2525.2	0.1705	0.04518	338.98	1135.20

Contd...(Table 5.18)

Run No.	U	T	T _{c av}	Q	U _{oc}	h _{ic}	K	μ ₂	τ _w g _c	$\frac{8U}{D_t}$
1	2	3	4	5	6	7	8	9	10	11
Agitator Diameter = 12.7 cm										
1	64.07	50.8	40.175	40.75	778.2	1289.5	0.1735	0.06593	43.97	270.05
2	81.85	50.3	40.100	40.75	860.3	1483.0	0.1740	0.06246	62.36	344.90
3	100.74	50.10	40.275	39.04	834.6	1429.4	0.1730	0.05880	83.94	424.60
4	117.76	50.0	40.450	39.88	930.3	1671.1	0.1728	0.05630	105.53	496.30
5	133.50	50.0	40.650	40.75	963.8	1770.8	0.1720	0.05469	121.52	562.69
6	150.40	50.1	41.025	42.90	1094.0	2137.6	0.1710	0.05215	152.70	633.90
7	178.39	50.3	41.425	42.90	1077.5	2119.1	0.1700	0.04998	187.88	751.90
8	216.64	51.7	43.450	40.75	1147.0	2327.2	0.1635	0.04578	239.85	913.13
9	240.19	52.1	43.775	40.75	1095.6	2157.9	0.1621	0.04409	278.23	1012.30
10	249.52	52.60	44.100	46.12	1165.6	2422.5	0.1615	0.04328	297.40	1051.70
11	267.59	53.0	44.375	49.33	1294.0	3125.0	0.1605	0.04221	332.60	1127.80

TABLE 5.19: HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL EXPERIMENTAL DATA

Jacket Fluid: Water

Coil Fluid: 2% CMC-A (n= 0.793)

Vessel Fluid: 2% CMC-A (n=0.793)

Run No.	U	T	T _{cav}	Q	U _{oc}	h _{ic}	K	μ ₂	τ _w g _c	$\frac{8U}{D_t}$
1	2	3	4	5	6	7	8	9	10	11
Agitator Diameter = 12.7 cm										
1	19.15	60.15	42.985	18.62	203.0	325.74	0.880	0.3608	27.15	80.70
2	36.63	58.70	42.575	21.90	324.0	658.84	0.890	0.2972	59.11	154.39
3	56.53	58.00	42.325	25.12	397.8	964.32	0.890	0.2572	103.84	238.27
4	76.15	57.18	41.945	29.57	383.0	891.74	0.900	0.2376	148.57	320.96
5	113.65	56.80	41.750	28.47	454.7	1303.61	0.900	0.2106	234.80	479.00
6	138.45	56.50	41.750	30.66	489.0	1561.03	0.900	0.1998	290.37	583.56
7	168.40	56.40	41.900	31.76	513.0	1797.90	0.900	0.1863	375.44	709.70
8	201.90	56.30	42.375	29.42	494.6	1625.22	0.890	0.1746	461.68	851.00
9	213.45	56.32	42.275	29.84	518.74	1853.90	0.890	0.1709	490.49	899.68

Contd....(Table 5.19)

Run No.	U	T	T _{cav}	Q	U _{oc}	h _{ic}	K	μ ₂	τ _w g _c	$\frac{8U}{D_t}$
1	2	3	4	5	6	7	8	9	10	11
Agitator Diameter = 12.7 cm										
1	19.15	61.40	43.410	22.57	248.5	387.50	0.871	0.3553	27.15	80.70
2	36.66	60.80	43.375	25.80	333.4	584.18	0.874	0.2919	59.11	154.50
3	67.50	59.90	43.350	26.87	375.0	697.60	0.874	0.2399	123.02	2854.50
4	95.19	59.25	43.225	29.45	446.5	933.88	0.875	0.2187	177.34	401.22
5	122.3	58.42	43.185	34.40	537.0	1235.60	0.877	0.2025	221.19	515.19
6	149.99	58.00	43.000	40.20	642.0	2011.26	0.880	0.1909	305.40	632.00
7	181.15	57.30	42.900	38.70	604.0	1719.69	0.880	0.1812	375.44	763.50
8	196.15	56.70	43.030	38.70	655.0	2119.09	0.880	0.1758	423.32	826.70
9	214.60	56.60	43.415	38.70	672.0	2282.58	0.871	0.1672	480.85	904.50

**TABLE 5.20: PRESSURE DROP TO FLUIDS FLOWING THROUGH HEAT TRANSFER TEST COIL CALCULATED
VARIABLES**

$D_t = 1.898 \text{ cm}$

$D_c = 34.0 \text{ cm}$

Run No.	U	$N_{Red} \times 10^4$	f_c	f_{SL_2}	f_c/f_{SL_2}	f_{STD}	$f_c(D_c/D_t)^{1/2}$	$f_{STD}(D_c/D_t)^{1/2}$	$N_{Red}(D_c/D_t)^{1/2}$
1	2	3	4	5	6	7	8	9	10
Fluid: Water (n=1.0); Agitator Diameter = 12.7 cm; R.P.M.= 100									
1	27.96	7.070	0.00730	-	-	0.00861	0.00309	0.0364	22.03
2	33.26	8.418	0.01050	-	-	0.00824	0.00444	0.0349	26.23
3	43.86	11.099	0.01000	-	-	0.00769	0.00423	0.0325	34.58
4	56.81	14.377	0.00976	-	-	0.00721	0.00410	0.0305	44.80
5	74.47	18.847	0.00780	-	-	0.00674	0.00330	0.0285	58.73
6	100.0	25.328	0.00870	-	-	0.00626	0.00368	0.0265	78.92
7	137.1	34.714	0.00836	-	-	0.00578	0.00354	0.0245	108.17
8	177.7	44.990	0.00738	-	-	0.00542	0.00312	0.0229	140.19
9	211.3	53.487	0.00787	-	-	0.00519	0.00333	0.0219	166.67
10	246.2	62.307	0.00771	-	-	0.00500	0.00326	0.0212	194.16
11	261.1	66.077	0.00849	-	-	0.00493	0.00359	0.0208	205.91
Fluid: Water (n=1.0); Agitator Diameter = 12.7 cm; R.P.M.= 300									
1	20.90	5.289	0.01180	-	-	0.00926	0.00499	0.0392	16.48
2	36.50	9.237	0.01027	-	-	0.00806	0.00434	0.0341	28.78
3	53.57	13.556	0.00970	-	-	0.00732	0.00410	0.0310	42.24
4	58.58	14.825	0.00944	-	-	0.00716	0.00399	0.0303	46.19

Contd... (Table 5.20)

Run No.	U	$N_{Red} \times 10^{-3}$	f_c	f_{SL_2}	f_c/f_{SL_2}	f_{STD}	$f_c(D_c/D_t)^{1/2}$	$f_{STD} (D_c/D_t)$	$N_{Red}(D_c/D_t)^{1/2}$
1	2	3	4	5	6	7	8	9	10
Fluid: Water (n=1.0); Agitator Diameter = 12.7 cm; R.P.M.= 300									
5	79.77	20.187	0.00682	-	-	0.00663	0.0288	0.0280	62.90
6	98.02	24.806	0.00901	-	-	0.00630	0.0381	0.0266	77.30
7	136.88	34.639	0.00840	-	-	0.00579	0.0355	0.0245	107.94
8	180.15	45.589	0.00768	-	-	0.00540	0.0325	0.0228	142.00
9	214.89	54.380	0.00761	-	-	0.00517	0.0322	0.0219	169.45
10	251.98	63.767	0.00735	-	-	0.00497	0.0311	0.0210	198.71
11	279.65	70.774	0.00737	-	-	0.00484	0.0312	0.0205	220.53
Fluid: Water (n=1.0); Agitator Diameter = 12.7 cm; R.P.M.= 800									
1	35.91	9.088	0.00980	-	-	0.00809	0.0414	0.0342	28.32
2	40.03	10.130	0.01590	-	-	0.00787	0.0672	0.0333	31.56
3	59.46	15.047	0.00916	-	-	0.00713	0.0387	0.0302	46.89
4	78.30	19.815	0.00910	-	-	0.00666	0.0380	0.0282	61.74
5	98.02	24.807	0.00891	-	-	0.00629	0.0377	0.0266	77.30
6	115.98	29.350	0.00871	-	-	0.00603	0.0368	0.0255	91.46
7	137.17	34.714	0.00764	-	-	0.00578	0.0323	0.0244	108.17
8	178.97	45.293	0.00758	-	-	0.00511	0.0321	0.0229	141.14
9	212.53	53.786	0.00757	-	-	0.00518	0.0320	0.0219	167.60
10	250.80	63.469	0.00747	-	-	0.00497	0.0316	0.0210	197.78
11	280.24	70.919	0.00749	-	-	0.00484	0.0317	0.0204	220.99

Contd... (Table 5.20)

Run No.	U	$N_{Red} \times 10^{-3}$	f_c	f_{SL_2}	f_c/f_{SL_2}	f_{STD}	$f_c(D_c)$	$f_{STD}(D_c)$	$N_{Red}(D_c/D_t)^{1/2}$
1	2	3	4	5	6	7	8	9	10
Fluid: 0.5% CMC-A (n=0.937); Agitator Diameter = 18.35 cm; R.P.M.= 150									
1	270.86	2.280	8.462	7.476	11.316	0.00642	0.0358	0.0271	71.050
2	261.44	2.177	8.381	7.843	10.684	0.00655	0.0354	0.0277	67.839
3	244.95	2.017	8.722	8.465	10.301	0.00663	0.0369	0.0280	62.850
4	233.76	1.910	8.551	8.938	9.565	0.00672	0.0361	0.0284	59.519
5	216.69	1.759	8.725	9.702	8.992	0.00686	0.0369	0.0290	54.810
6	198.73	1.590	8.995	10.738	8.376	0.00730	0.0381	0.0297	49.540
7	173.11	1.366	9.237	10.502	7.389	0.00730	0.0391	0.0310	42.560
8	147.50	1.118	9.782	15.267	6.407	0.00768	0.0414	0.0325	34.839
9	114.82	0.861	10.195	19.826	5.142	0.00820	0.0431	0.0347	26.830
10	85.38	0.618	11.415	27.580	4.139	0.00891	0.0483	0.0377	19.258
11	57.70	0.407	13.456	41.881	3.213	0.00989	0.0569	0.0418	12.680

Contd... (Table 5.20)

Run No.	U	$N_{Red} X 10$	$f_c X 10^2$	$f_{SL_2} X 10^2$	f_c / f_{SL_2}	f_{STD}	$f_c (D_c / D_t)^{1/2}$	f_{STD}	$N_{Red} (D_c / D_t)^{1/2}$
1	2	3	4	5	6	7	8	9	10
Fluid: 1% CMC-A (n=0.851); Agitator Diameter = 12.7 cm; R.P.M.=									
1	29.84	0.857	3.199	2.191	1.460	0.01461	0.1354	0.0618	2.670
2	46.96	1.504	2.224	1.251	1.779	0.01268	0.0941	0.0536	4.686
3	64.13	2.240	2.078	0.842	2.467	0.01148	0.0879	0.0485	6.980
4	100.85	3.913	1.649	0.481	3.437	0.00998	0.0698	0.0422	12.193
5	137.10	5.700	1.366	0.329	4.151	0.00909	0.0578	0.0384	17.760
6	164.40	7.156	1.206	0.262	4.605	0.00858	0.0510	0.0363	22.299
7	188.89	8.550	1.118	0.219	5.105	0.00821	0.0473	0.0347	26.641
8	122.70	10.610	1.015	0.177	5.733	0.00778	0.0429	0.0329	33.062
9	243.69	11.759	0.954	0.159	5.971	0.00758	0.0404	0.0320	36.641
10	269.34	13.419	0.925	0.140	6.608	0.00734	0.0391	0.0310	41.816

Contd... (Table 5.20)

Run No.	U	$N_{Red} \times 10^{-3}$	$f_c \times 10^2$	$f_{SL_2} \times 10$	f_c / f_{SL_2}	f_{STD}	$f_c (D_c / D_t)^{1/2}$	$f_{STD} (L$	$N_{Red} (D_c / D_t)^{1/2}$
1	2	3	4	5	6	7	8	9	10
Fluid: 1% CMC-A(n=0.851); Agitator Diameter = 12.7 cm; R.P.M.=									
1	64.07	2.188	2.120	0.859	2.468	0.01155	0.0897	0.0489	6.818
2	81.85	2.952	1.843	0.636	2.898	0.01070	0.0780	0.0453	9.190
3	100.74	3.859	1.637	0.487	3.361	0.01000	0.0693	0.0423	12.021
4	117.76	4.710	1.507	0.399	3.776	0.00953	0.0637	0.0403	14.614
5	133.50	5.498	1.351	0.341	3.948	0.00917	0.0571	0.0388	17.133
6	150.40	6.496	1.336	0.289	4.622	0.00881	0.0565	0.0372	20.241
7	178.39	7.990	1.168	0.235	4.970	0.00835	0.0494	0.0353	24.894
8	216.64	10.660	1.012	0.176	5.749	0.00777	0.0429	0.0329	33.218
9	240.19	12.270	0.955	0.153	6.241	0.00751	0.0404	0.0317	38.235
10	249.52	12.980	0.945	0.144	6.535	0.00740	0.0400	0.0313	40.448
11	267.59	14.280	0.916	0.131	6.960	0.00722	0.0387	0.0305	44.490

Contd... (Table 5.20)

Run No.	U	$N_{Red} X 10^3$	$f_c X 10^3$	$f_{SL_2} X 10^3$	f_c / f_{SL_2}	f_{STD}	$f_c (D_c / D_t)^{1/2}$	$f_{STD} (D)$	$N_{Red} (D_c / D_t)^{1/2}$
1	2	3	4	5	6	7	8	9	10
Fluid: 2% CMC-A(n=0.793); Agitator Diameter = 12.7 cm; R.P.M.=									
1	213.4 5	3.013	21.09	6.694	3.150	-	-	-	-
2	201.9 3	2.770	22.19	7.271	3.052	-	-	-	-
3	168.4 1	2.180	25.94	9.248	2.805	-	-	-	-
4	138.4 5	1.674	29.68	12.048	2.463	-	-	-	-
5	113.6 5	1.303	35.62	15.471	2.302	-	-	-	-
6	76.15	0.774	50.21	26.058	1.926	-	-	-	-
7	56.53	0.530	63.67	38.102	1.675	-	-	-	-
8	36.63	0.297	86.32	67.790	1.273	-	-	-	-
9	19.15	0.128	145.06	157.328	0.922	-	-	-	-
Fluid: 2% CMC-A(n=0.793); Agitator Diameter = 12.7 cm; R.P.M.=									
1	36.66	0.302	86.32	66.660	1.295	-	-	-	-
2	67.50	0.679	52.51	29.680	1.782	-	-	-	-
3	95.19	1.050	38.35	19.207	1.996	-	-	-	-
4	122.3 4	1.459	31.69	13.828	2.212	-	-	-	-
5	149.9 9	1.899	26.61	10.623	2.504	-	-	-	-
6	181.8 5	2.416	22.41	8.356	2.683	-	-	-	-
7	196.1 5	2.696	21.55	7.485	2.881	-	-	-	-
8	214.6 2	3.102	20.45	6.504	3.145	-	-	-	-

Note: Columns indicated by dash (-) are in turbulent region

TABLE 5.21: LAMINAR FLOW HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL CALCULATED

VARIABLES

Jacket Fluid: Water: (22.4 kg/min)

Coil Fluid: Water (n= 1.0)

Tank Fluid: Water (n=1.0)

Run No.	U	N_{Nuic}	$N_{Re_2} \times 10$	$N_{D_2} \times 10^{-3}$	N_{Gz}	N_{Pr_2}	N_{Nuis}	$\frac{N_{Nuic}}{N_{Nuis}}$	$\frac{N_{Nuic}}{N_{Nuis}} - 1$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{D_2}^{-1/2}$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{Pr_2}^{-0.12}$
1	2	3	4	5	6	7	8	9	10	11	12
Agitator Diameter = 12.7 cm; R.P.M.= 100											
1	27.965	65.25	7.070	1.670	143.060	4.966	11.370	5.738	4.738	0.1159	3.91
2	33.263	52.76	8.418	1.989	170.160	4.966	12.390	4.258	3.258	0.0730	2.68
3	43.860	77.45	11.099	2.622	224.376	4.966	14.100	5.490	4.490	0.0876	3.70
4	56.810	123.76	14.377	-	-	4.966	-	-	-	-	-
5	74.476	224.38	18.847	-	-	4.966	-	-	-	-	-
6	100.085	193.00	25.328	-	-	4.966	-	-	-	-	-
7	137.176	184.40	34.714	-	-	4.966	-	-	-	-	-
8	177.790	361.81	44.990	-	-	4.966	-	-	-	-	-
9	211.357	361.81	53.487	-	-	4.966	-	-	-	-	-
10	246.210	393.10	62.307	-	-	4.966	-	-	-	-	-
11	261.106	438.60	66.077	-	-	4.966	-	-	-	-	-

Contd... (Table 5. 21)

TABLE 5. 21: LAMINAR FLOW HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL CALCULATED VARIABLES

Jacket Fluid: Water: (22.4 kg/min)

Coil Fluid: Water (n= 1.0)

Tank Fluid: Water (n=1.0)

Run No.	U	N_{Nuic}	$N_{Re_2} \times 10^{-3}$	$N_{D_2} \times 10^{-3}$	N_{Gz}	N_{Pr_2}	N_{Nuis}	$\frac{N_{Nuic}}{N_{Nuis}}$	$\frac{N_{Nuic}}{N_{Nuis} - 1}$	$\left(\frac{N_{Nuic}}{N_{Nuis} - 1}\right) N$	$\left(\frac{N_{Nuic}}{N_{Nuis} - 1}\right) N_{Pr_2}^{-0.12}$
1	2	3	4	5	6	7	8	9	10	11	12
Agitator Diameter = 12.7 cm; R.P.M.= 300											
1	20.900	33.80	5.289	1.249	106.917	4.966	8.294	4.074	3.074	0.0870	2.53
2	36.500	55.60	9.237	2.182	186.732	4.966	9.975	3.569	2.569	0.0550	2.12
3	53.570	85.45	13.556	3.202	274.070	4.966	11.370	7.515	6.515	0.1150	5.37
4	58.580	113.58	14.825	-	-	4.966	-	-	-	-	-
5	79.770	145.53	20.187	-	-	4.966	-	-	-	-	-
6	98.025	184.44	24.806	-	-	4.966	-	-	-	-	-
7	136.880	286.65	34.639	-	-	4.966	-	-	-	-	-
8	180.150	361.80	45.589	-	-	4.966	-	-	-	-	-
9	214.890	490.37	54.380	-	-	4.966	-	-	-	-	-
10	251.980	931.06	63.767	-	-	4.966	-	-	-	-	-
11	279.650	555.97	70.770	-	-	4.966	-	-	-	-	-

Contd... (Table 5. 21)

TABLE 5.21: LAMINAR FLOW HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL CALCULATED VARIABLES

Jacket Fluid: Water: (22.4 kg/min)

Coil Fluid: Water (n= 1.0)

Tank Fluid: Water (n=1.0)

Run No.	U	N_{Nuic}	$N_{Re_2} \times 10^{-3}$	$N_{D_2} \times 10^{-3}$	N_{Gz}	N_{Pr_2}	N_{Nuis}	$\frac{N_{Nu}}{N_{Nu}}$	$\frac{N_{Nuic}}{N_{Nuis} - 1}$	$\left(\frac{N_{Nuic}}{N_{Nuis} - 1}\right) N_{D_2}^{-1/2}$	$\left(\frac{N_{Nuic}}{N_{Nuis} - 1}\right) N$
1	2	3	4	5	6	7	8	9	10	11	12
Agitator Diameter = 12.7 cm; R.P.M.= 800											
1	35.913	63.25	9.088	2.147	183.710	4.966	9.920	6.376	5.376	0.1160	4.43
2	40.030	66.22	10.130	-	-	4.966	-	-	-	-	-
3	59.460	150.82	15.047	-	-	4.966	-	-	-	-	-
4	78.300	184.44	19.815	-	-	4.966	-	-	-	-	-
5	98.025	212.89	24.807	-	-	4.966	-	-	-	-	-
6	115.980	251.73	29.350	-	-	4.966	-	-	-	-	-
7	137.176	268.16	34.714	-	-	4.966	-	-	-	-	-
8	178.977	328.00	45.293	-	-	4.966	-	-	-	-	-
9	212.535	1203.96	53.786	-	-	4.966	-	-	-	-	-
10	250.803	2885.50	63.469	-	-	4.966	-	-	-	-	-
11	280.240	1203.96	70.919	-	-	4.966	-	-	-	-	-

TABLE 5.22: LAMINAR FLOW HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL CALCULATED VARIABLES

Jacket Fluid: Water: (21.318 kg/min)

Coil Fluid: 0.5% CMC (n= 0.937)

Tank Fluid: 4% CMC (n= 0.698)

Run No.	U	N_{Nuic}	$N_{Re_2} \times 10^{-3}$	$N_{D_2} \times 10^{-3}$	N_{Gz}	N_{Pr_2}	N_{Nuis}	$\frac{N_{Nuic}}{N_{Nuis}}$	$\frac{N_{Nuic}}{N_{Nuis} - 1}$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_i$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_j$
1	2	3	4	5	6	7	8	9	10	11	12
Agitator Diameter = 18.35 cm											
1	270.86	115.420	2.141	5.056	1385.4	15.87	19.674	5.866	4.866	0.0683	2.494
2	261.44	83.012	2.042	4.819	1337.2	16.08	19.392	4.280	3.280	0.0472	2.351
3	244.95	94.360	1.890	4.460	1252.8	16.20	18.829	5.011	4.011	0.0606	2.870
4	233.76	106.770	1.790	4.229	1195.6	16.38	18.548	5.756	4.456	0.0730	3.402
5	216.69	103.850	1.649	3.890	1108.3	16.50	18.300	5.674	4.674	0.0749	3.338
6	198.73	98.909	1.490	3.520	1016.4	16.74	17.700	5.588	4.588	0.0773	3.272
7	173.12	122.250	1.280	3.020	885.4	16.96	16.894	7.236	6.236	0.1134	4.440
8	147.50	88.325	1.048	2.476	754.4	17.67	15.978	5.527	4.527	0.0908	3.304
9	114.82	57.710	0.807	1.906	587.3	17.87	14.747	3.913	2.913	0.0666	2.057
10	85.38	58.410	0.580	1.370	436.7	18.41	13.300	4.390	3.390	0.0916	2.916
11	57.70	51.190	0.382	0.902	295.1	18.94	11.702	4.374	3.374	0.1123	2.258

TABLE 5.23: LAMINAR FLOW HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL CALCULATED VARIABLES

Jacket Fluid: Water: (22.4 kg/min)

Coil Fluid: 1% CMC (n= 0.851)

Tank Fluid: 1% CMC (n= 0.851)

Run No.	U	N_{Nuic}	$N_{Re_2} \times 10^{-3}$	$N_{D_2} \times 10^{-1}$	N_{Gz}	N_{Pr_2}	N_{Nuis}	$\frac{N_{Nuic}}{N_{Nuis}}$	$\frac{N_{Nuic}}{N_{Nuis}} - 1$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{D_2}^{-1/2}$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{Pr_2}^{-0.1}$
1	2	3	4	5	6	7	8	9	10	11	12
Agitator Diameter = 12.7 cm											
1	29.84	25.559	0.730	0.172	154.20	51.78	9.515	2.686	1.686	0.1286	1.050
2	46.96	44.530	1.280	0.302	242.59	46.50	11.059	4.026	3.026	0.1742	1.909
3	64.13	48.158	1.900	0.448	331.29	42.40	12.248	3.932	2.932	0.1385	1.869
4	100.85	50.510	3.330	0.786	521.03	38.38	14.201	3.556	2.556	0.0912	1.650
5	137.00	64.729	4.860	1.148	707.65	35.76	15.799	4.102	3.102	0.0915	2.018
6	164.40	65.750	6.090	1.438	849.30	34.24	16.793	3.915	2.195	0.0768	1.909
7	188.89	88.169	7.280	1.720	975.80	32.90	17.574	5.017	4.017	0.0968	2.641
8	122.70	101.080	9.030	2.132	1150.49	31.29	18.639	5.423	4.423	0.0957	2.923
9	243.69	129.550	10.007	2.364	1258.90	30.90	18.994	6.820	5.820	0.1197	3.856
10	269.34	88.169	11.420	2.698	1391.40	29.92	19.846	4.442	3.442	0.0662	2.290

Contd... (Table 5.23)

Run No.	U	N_{Nuic}	$N_{Re_2} \times 10$	$N_{D_2} \times 10$	N_{Gz}	N_{Pr_2}	N_{Nuis}	$\frac{N_{Nuic}}{N_{Nuis}}$	$\frac{N_{Nuic}}{N_{Nuis}} - 1$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{D_2}^{-1/2}$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{Pr_2}^{-0.1}$
1	2	3	4	5	6	7	8	9	10	11	12
Agitator Diameter = 12.7 cm											
1	267.59	109.100	12.150	2.870	1382.40	27.95	19.882	5.487	4.487	0.0837	3.007
2	249.52	84.580	11.050	2.610	1289.00	28.66	19.172	4.411	3.411	0.0667	2.280
3	240.19	75.340	10.440	2.466	1240.80	29.19	19.118	3.940	2.940	0.0592	1.962
4	216.64	81.250	9.070	2.142	119.17	30.30	18.462	4.400	3.400	0.0734	2.259
5	178.39	73.988	6.800	1.606	921.60	33.09	17.219	4.296	3.296	0.0822	2.165
6	150.40	74.636	5.528	1.306	777.03	34.53	16.330	4.570	3.570	0.0988	2.336
7	133.50	61.829	4.679	1.105	689.69	36.20	15.620	3.958	2.958	0.0889	1.923
8	117.76	58.347	4.008	0.946	608.37	37.28	15.089	3.866	2.866	0.0932	1.857
9	100.74	49.900	3.284	0.775	520.43	38.94	14.200	3.514	2.514	0.0902	1.615
10	81.85	51.779	2.512	0.593	422.85	41.36	13.314	3.889	2.889	0.1186	1.848
11	64.07	45.020	1.861	0.439	330.99	43.66	12.248	4.397	3.397	0.1276	1.701

TABLE 5. 24: LAMINAR FLOW HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL CALCULATED VARIABLES

Jacket Fluid: Water: (21.318 kg/min)

Coil Fluid: 2% CMC (n= 0.793)

Tank Fluid: 2% CMC (n= 0.793)

Run No.	U	N_{Nuic}	$N_{Re_2} \times 10$	$N_{D_2} \times 10^{-3}$	N_{Gz}	N_{Pr_2}	N_{Nuis}	$\frac{N_{Nuic}}{N_{Nuis}}$	$\frac{N_{Nuic}}{N_{Nuis}} - 1$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{D_2}^{-1/2}$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{Pr_2}^{-0.12}$
1	2	3	4	5	6	7	8	9	10	11	12
Agitator Diameter = 12.7 cm											
1	213.45	64.728	2.39	0.5646	1114.35	113.16	18.587	3.482	2.482	0.1045	1.407
2	201.90	56.740	2.20	0.5197	1054.10	115.60	18.300	3.100	2.100	0.0921	1.188
3	168.40	62.770	1.73	0.4087	879.40	123.37	17.068	3.677	2.677	0.1325	1.502
4	138.45	54.500	1.33	0.3137	722.80	132.30	15.996	3.407	2.407	0.1360	1.339
5	113.65	45.500	1.03	0.2440	593.30	139.47	14.995	3.034	2.034	0.1302	1.123
6	76.15	31.240	0.61	0.1450	397.50	157.35	13.082	2.388	1.388	0.1152	0.757
7	56.53	33.669	0.42	0.0994	295.10	170.33	11.867	2.837	1.837	0.1842	0.990
8	36.63	22.990	0.24	0.0557	191.20	196.82	10.276	2.237	1.237	0.1658	0.656
9	19.15	11.370	0.10	0.0240	99.99	238.90	8.290	1.371	0.371	0.0757	0.192

Contd... (Table5.24)

Run No.	U	N_{Nuic}	$N_{Re_2} \times 10^1$	$N_{D_2} \times 10^{-3}$	N_{Gz}	N_{Pr_2}	N_{Nuis}	$\frac{N_{Nuic}}{N_{Nuis}}$	$\frac{N_{Nuic}}{N_{Nuis} - 1}$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_D^-$	$\left(\frac{N_{Nuic}}{N_{Nuis}} - 1\right) N_{Pr_2}^{-0.1}$
1	2	3	4	5	6	7	8	9	10	11	12
Agitator Diameter = 12.7 cm											
1	19.15	13.529	0.10	0.0244	99.99	235.29	8.290	1.6319	0.6319	0.128	0.328
2	36.66	20.396	0.24	0.0567	191.36	193.30	10.276	1.9848	0.9848	0.131	0.523
3	67.49	24.350	0.54	0.1273	352.37	158.87	12.600	1.9320	0.9320	0.082	0.507
4	95.19	32.600	0.83	0.1968	496.90	144.83	14.119	2.3080	1.3080	0.093	0.719
5	122.30	43.140	1.16	0.2730	638.49	134.10	15.370	2.8060	1.8060	0.109	1.003
6	149.99	70.220	1.50	0.3558	783.05	126.40	16.407	4.2790	3.2790	0.174	1.834
7	181.15	60.040	1.92	0.4526	945.69	119.99	17.515	3.4270	2.4270	0.114	1.365
8	196.15	73.988	2.14	0.5050	1024.00	116.42	17.997	4.1110	3.1110	0.138	1.757
9	214.60	79.696	2.46	0.5810	1120.37	110.72	18.605	4.2830	3.2830	0.136	1.865

TABLE 5.25: COMPARISON BETWEEN EXPERIMENTAL AND CALCULATED NUSSELT NUMBERS FOR LAMINAR HEAT TRANSFER TO FLUIDS FLOWING THROUGH COIL

S. No.	Fluid	N_{Nuic} (Experimental)	N_{Nuic} (Experimental)
1	2	3	4
1	0.5% CMC-A	115.42	Turbulant
		83.012	Turbulant
		94.36	Turbulant
		106.77	Turbulant
		103.85	Turbulant
		98.90	Turbulant
		122.25	Turbulant
		88.325	Turbulant
		57.71	74.9
		58.41	59.4
	51.19	44.705	
2	1% CMC-A	25.559	22.73
		44.53	31.14
		48.158	39.06
		50.51	54.89
		64.729	69.90
		65.75	80.90
		88.169	90.73
		101.08	104.6
		129.55	Turbulant
		88.169	Turbulant
		109.1	Turbulant
		84.58	Turbulant
		75.34	Turbulant
		81.25	Turbulant
		73.988	86.52
		74.636	75.83
		61.829	68.3
		58.847	62.3
49.9	54.7		
51.779	46.7		
	45.02	38.87	
3	2% CMC-A	64.728	69.95
		56.74	66.90
		62.77	57.60
		54.50	49.55
		45.50	42.975
		31.24	32.13
		33.669	26.32
		22.99	19.80
		11.37	13.44
			13.529

		20.396	19.888
		24.35	29.80
		32.60	37.87
		43.14	45.53
		70.22	52.32
		60.04	61.22
		73.988	65.23
		79.696	70.68

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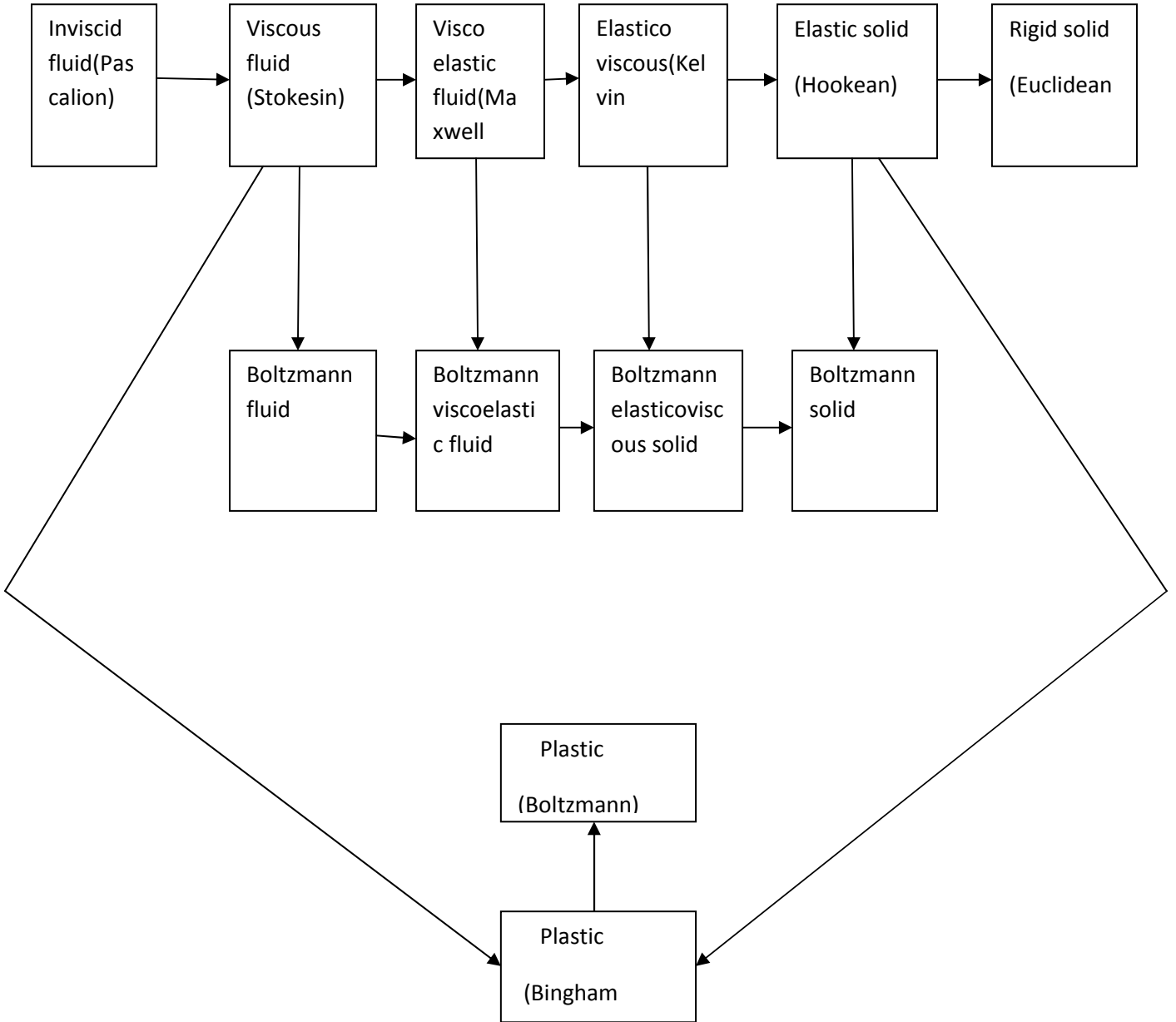
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APPENDIX – A

- CMC – A Carboxy methyl e cellulose manufactured by DANPHA CHEMICALS, india
- CMC – B Low viscosity grade carboxy methyl cellulose manufactured by CELLULOSE PRODUCT OF
INDIA LTD., India
- CMC – C High viscosity grade carboxy methyl e cellulose manufactured by CELLULOSE PRODUCT OF
INDIA LTD.,India
- PVA Polyvinyl accohol manufactured by CELLULOSE PRODUCT OF INDIA LTD., India.

APPENDIX – B1

Rheological classification of materials presented by Fredrickson (54):



APPENDIX – B2

Rheological classification of fluids presented by Metzner (25)

Category 1: Purely viscous or Time-Independent Fluids. The deformation rate depends upon the stress and is a single value function of the latter,

$$P_{ij} = \mu d_{ije}$$

$$\mu = \mu (I, II, III)$$

Special cases:

(a) **Newtonian:** The viscosity is independent of shear stress.

(b) **Non-Newtonian:** the viscosity is a function of shear stress.

(x), $\mu =$ decreasing function of the invariants of strain rate tensor I,II,III includes materials commonly termed “Pseudoplastic “, “Bingham plastic” or “Shear thinning”

(y), $\mu =$ increasing function of I,II,III “dilatants” or “shear thickening”.

Category 2: Time dependent fluids

The deformation rate and viscosity depend upon both the stress and duration of stress.

$$P_{ij} = \mu d_{ije}$$

$$\mu = \mu (I, II, III, \theta),$$

Category 3: viscoelastic Fluids

The deformation rate and viscosity depend upon both the stress and the extent of deformation.

Category 4: More complex Materials

Fluids exhibiting the characteristics of more than one of categories 1, 2 and 3.

APPENDIX – B3

Table Models for Non-Newtonian fluids.

1. Two parameter Ostwald de Wacle, or power law (4)

$$\tau = -K \left(\frac{du}{dy} \right)^n$$

2. Generalized Power law

$$\tau_w = -K^t \left(\frac{8U}{D} \right)^n$$

3. Bingham equation (2)

$$\tau - \tau_y = -P \left(\frac{du}{dy} \right)$$

4. Ellis Model ()

$$\left(\frac{du}{dy} \right) = - \left[\left(\frac{1}{u_0} \right) - K \tau^{1/(n-1)} \right]$$

5. Sisko model (36)

$$\tau = \left[\mu_\infty + K \left(\frac{du}{dy} \right)^{n-1} \right] \left(\frac{du}{dy} \right)$$

6. Prandtl model

$$\tau = A \sin h^{-1} \left[\frac{1}{C} \left(\frac{du}{dy} \right) \right]$$

7. Eyring Model (10)

$$\tau = \frac{1}{B} \left(\frac{du}{dy} \right) + C \sin \left(\frac{\tau}{A} \right)$$

8. Reiner- Philippoff Model (17)

$$= \left[\mu_\infty + \frac{\mu_0 - \mu_\infty}{1 + \frac{\tau^2}{A}} \right] \left(\frac{du}{dy} \right)$$

9. Three parameter Powell-Eyring Model (10)

$$\mu = \mu_\infty + \left[\frac{\alpha_1}{(du/dy)} \right] \sin h^{-1} [\alpha_2 (du/dy)]$$

10. Five Parameter Powel-Eyring Model (10)

$$\mu = \mu_\infty \left[\frac{\alpha_1}{(du/dy)} \right] \sin h^{-1} [\alpha_2 (du/dy)] + \frac{\alpha_3}{(du/dy)} \sin h^{-1} [\alpha_4 (du/dy)]$$

11. Ree- Model (37)

$$\mu = \mu_\infty + (\mu_0 - \mu_\infty) e^{-A\tau}$$

12. Williamson Model (45) $\tau = \frac{A \left(\frac{du}{dy} \right)}{B + \left(\frac{du}{dy} \right)} + \mu_\infty \left(\frac{du}{dy} \right)$

13. Herschel- Bulkley Law

$$\tau - \tau_y = K (du/dy)^{n-1} (du/dy)$$

14. Casson Law (32)

$$\tau^{1/2} - \tau_y^{1/2} = -K (du/dy)^{1/2}$$

15. Heinz (30)

$$\tau^{2/3} - \tau_y^{2/3} = K(du/dy)^{2/3}$$

16. Bruss Model

$$\tau^n - \tau_y^n = -K(du/dy)^{n-1} \left(\frac{du}{dy} \right)$$

17. Oldroyd Model (19)

$$\mu = \mu_0 \frac{[1 + \alpha_1 (\frac{du}{dy})^2]}{[1 + \alpha_2 (\frac{du}{dy})^2]}$$

18. Sprigg-Bird (55)

$$\mu = \mu_0 + \frac{3(\mu_0 - \mu_\infty)}{\sqrt{\pi^2 \alpha_1} (du/dy)} \left[\frac{\sin h\pi \sqrt{\alpha_1} (du/dy) - \sin \pi \sqrt{\alpha_1} (du/dy)}{\cos h\pi \sqrt{\alpha_1} (du/dy) - \cos \pi \sqrt{\alpha_1} (du/dy)} \right]$$

19. Bueche Model (23)

$$\mu = \mu + (\mu_0 - \mu_\infty) \left[\frac{2 + \alpha_1 (du/dy) - \sqrt{4\alpha_1 (\frac{du}{dy}) + \alpha_1^2 (du/dy)^2}}{2} \right]$$

20. Extended Williamson Model (45)

$$\mu = \mu_\infty + (\mu_0 - \mu_\infty) / \left[1 + \left(\frac{du/dy}{\alpha_1} \right)^2 \right]$$

21. Meter Model (50)

$$\mu = \mu_\infty + \left[\frac{\mu_0 - \mu_\infty}{1 + (\tau/\tau_m)^2} \right]$$

22. Crowley-Kitzes (33)

$$\tau_{yx} = \frac{\mu L}{g_c} \left[\frac{1.2 + S (C\tau_{yx}^{-0.2} + 1)^3}{1.2 - 2S (C\tau_{yx}^{-0.2} + 1)^3} \right] \left(\frac{du}{dy} \right)$$

23. De Haven (40)

$$\tau_{yx} = \frac{\mu_0/\mu_c}{1 + C \tau_{yx}^n} (du/dy)$$

24. Symonds et al. (24)

$$(\tau_{rx})_{r=R} = \tau_W = a(8U/D)^{1-K}$$

25. Spencer-Dillon (16)

$$\tau_W = \frac{8U/D}{(g_c/\mu_\infty) \left[1 + K \tau_W + \frac{(K\tau_W)^2}{2!} + \frac{(K\tau_W)^3}{3!} + \frac{(K\tau_W)^4}{4!} \right]}$$

26. Reiner-Rivlin (17)

$$\tau_W = \frac{1}{g_c} \left[(\mu_0 - \mu_\infty)^{-\tau_W F/4\mu} + \mu_\infty \right] + \left(\frac{8U}{D} \right)$$

SAMPLE CALCULATIONS

Table 9, Run No. 1

Coil fluid 2% CMC-A

Vessel fluid 2% CMC-A

Inner diameter of the coil tube, $D_t = 1.898$ cm

Outer diameter of the coil tube, $D_o = 2.25$ cm

Effective length of the coil, $L = 366.48$ cm

Diameter of the agitated vessel, $D_T = 45.25$ cm

Height of the fluid level in the vessel, $H = 44.0$ cm

Inlet bulk temperature of water flowing through the jacket at steady state condition, $T_{ji} = 70.8^{\circ}\text{C}$

Outlet bulk temperature of water flowing through the jacket at steady state condition, $T_{jo} = 70.1^{\circ}\text{C}$

Inlet bulk temperature of the fluid flowing through the coil, $T_{ci} = 40.42^{\circ}\text{C}$

Outlet bulk temperature of the fluid flowing through the coil, $T_{co} = 39.79^{\circ}\text{C}$

Mass rate flow of water flowing through the jacket, $W_j = 20.52$ kg/min

Mass rate flow of fluid flowing through the coil, $W_c = 22.5$ kg/min

Temperature of the fluid in the agitated vessel, $T = 55^{\circ}\text{C}$

RPM of the agitator, $N = 50$

Diameter of the agitator blade, $D_a = 7.5$ cm

Number of blades in the agitator = 4

Rate of heat transfer from the vessel wall to the agitated fluid: Q_j

$$Q_j = W_j \times C_p \times (T_{ji} - T_{jo}) = 20.52 \times 1.0 \times (70.8 - 70.1) = 14.364 \text{ Kcal/min.}$$

Rate of heat transfer from vessel fluid to the coil fluid: Q_c

$$Q_c = W_c \times c_p \times (T_{ci} - T_{co}) = 22.5 \times 1.0 \times (40.42 - 39.79) = 14.175 \text{ Kcal/min.}$$

Heat transfer coefficient for vessel wall to the agitated fluid

Temperature Driving Force: Δt_j

Since the ratio of the temperature differences is found to be less than 1.5, therefore the following arithmetic mean temperature difference method which is simpler and equally accurate for the calculation of temperature driving force is adopted.

$$\Delta t_j = T_{j,av} - T = \frac{T_{ji} - T_{jo}}{2} - T$$

$$= \frac{70.8 + 70.1}{2} - 55 = 15.45^\circ C$$

Area of the jacketed vessel surface available for heat transfer to the agitated fluid : A_j

$$A_j = \pi D_T \times H = \pi \times 45.25 \times 44.0 = 0.62574 \text{ m}^2$$

Overall heat transfer coefficient : U_j

$$U_j = \frac{Q}{A_j \times \Delta t_j} = \frac{14.364 \times 60}{0.62574 \times 15.45} = 89.2 \frac{\text{Kcal}}{\text{hr}} \text{ m}^2 \text{ }^\circ C$$

From the modified Wilson plot (Fig. 12) for 2% CMC-A, the intercept is obtained to be 0.00178

$$\frac{1}{U_j} = \frac{1}{h_j} + 0.000178$$

$$\text{Or } \frac{1}{89.2} = \frac{1}{h_j} + 0.000178$$

$$\text{Or } h_j = 105.93 \frac{\text{Kcal}}{\text{hrm}^2} \text{ }^\circ C$$

Heat Transfer Coefficient for Vessel Fluid to the Coil

Temperature Difference : Δt_c

$$(\Delta t_c)_{1m} = \frac{T_{ci} - T_{co}}{2.303 \log \frac{T_{ci} - T}{T_{co} - T}}$$

$$= \frac{40.42 - 39.79}{2.303 \log \frac{55 - 40.42}{55 - 39.79}} = 14.65^\circ C$$

Arithmetic Mean Temperature Driving Force : $(\Delta t_c)_m$

$$(\Delta t_c)_m = T - T_{c,av} = T - \frac{T_{ci} + T_{co}}{2} = 55 - \frac{40.42 + 39.79}{2} = 14.94^\circ C$$

Outer surface area of the test section of the coil available for heat transfer : A_c

$$A_c = \pi \times D_o \times L = \pi \times 2.25 \times 366.48 = 0.259 \text{ m}^2$$

Over all heat transfer coefficient : U_{oc}

$$U_{oc} = \frac{Q}{A_c \times \Delta t_c} = \frac{14.364 \times 60}{0.259 \times 14.94} = 223 \frac{\text{Kcal}}{\text{hrm}^2} \text{ }^\circ C$$

Film heat transfer coefficient at the outer surface of the coil : h_{oc}

From the Wilson plot (Fig. 12) for 2% CMC-A the value of the intercept is found to be 0.000905

$$\frac{1}{U_{oc}} = \frac{1}{h_{oc}} + 0.000905$$

$$\frac{1}{223} = \frac{1}{h_{oc}} + 0.000905$$

$$h_{oc} = 280.11 \frac{Kcal}{hrm^2} 0 c$$

Viscosity and dimensionless variables

Apparent viscosity : μ_a'' (defined by equation III.13)

From the plot of flow consistency index versus temperature (Fig.7) the value of K at the temperature of the stirred fluid, i.e., 55⁰c, has been obtained. From Fig. 7

$$K = 0.675 \text{ gm sec}^{n-2} / \text{cm}$$

Now,

$$\mu_a'' = K n^{1-n} (4\pi n)^{n-1}$$

Where, n = flow behavior index of the fluid = 0.793

$$\mu_a'' = 0.675 \times (0.793)^{1-0.793} (4 \times \pi \times \frac{50}{60})^{(0.793-1)}$$

$$= 0.39565 \text{ gm/sec cm}$$

Differential viscosity : μ_d'' (defined by equation III.17)

$$\mu_d'' = K n^{2-n} (4\pi N)^{n-1} = n\mu_a''$$

$$= 0.793 \times 0.39565 = 0.3137 \text{ gm/sec cm}$$

Nusselt Number : N_{Nuj} (experimental)

$$N_{Nuj} = \frac{h_j \times D_T}{k} = \frac{105.93 \times 45.25}{15.1 \times 10^{-4} \times 36000} = 88.17$$

N_{Nuj} = calculated from equation (IX.19) : N_{Nuj} (calculated)

$$N_{Nuj}(\text{calculated}) = 0.302(N_{Rea}'')^{2/3} (N_{Pra}'')^{1/3} \left(\frac{D_a}{D_T}\right)^{0.1} = 0.302 (379)^{2/3} (262)^{1/3} \left(\frac{7.5}{45.25}\right)^{0.1}$$

$$= 83.83$$

Reynolds Number : N_{Rea}'' (defined by equation VII.14)

$$N_{Rea}'' = \frac{D_a^2 \pi N \rho}{\mu_a''} = \frac{(7.5)^2 \times \pi \times 50 \times 1.02}{0.39565} = 379$$

Prandtl Number : N_{Pra}'' (defined by equation VII.19)

$$N_{Pra}'' = \frac{C_p \mu_a''}{k} = \frac{1 \times 0.39565}{15.1 \times 10^{-4}} = 262$$

Nusselt Number at the outer surface of the coil : N_{Nuoc}

$$N_{Nuoc} = \frac{h_{oc} \times D_0}{k} = \frac{280.11 \times 2.25}{15.1 \times 10^{-4} \times 36000} = 11.593$$

N_{Nuoc} calculated from equation (IX.21) : N_{Nuoc} (calculated)

$$N_{Nuoc} \text{ (calculated)} = 0.036 (N''_{Rea})^{2/3} (N''_{Pra})^{1/3} \left(\frac{D_a}{D_c}\right)^{0.1}$$

$$= 0.036 (379)^{2/3} (262)^{1/3} \left(\frac{7.5}{34.0}\right)^{0.1} = 9.968 \frac{N_{Nu_j}}{(N''_{Rea})^{1/3}} = \frac{88.18}{(262)^{1/3}} = 13.798$$

$$\frac{N_{Nu_j}}{(N''_{Rea})^{2/3} (N''_{Pra})^{1/3}} = \frac{88.17}{(379)^{2/3} (262)^{1/3}} = 0.26534 = \frac{88.17}{(262)^{1/3} (7.5/45.25)^{0.1}} = 16.516$$

$$\frac{N_{Nuoc}}{(N''_{Rea})^{2/3}} = \frac{11.593}{(379)^{2/3}} = 0.2229$$

$$\frac{N_{Nuoc}}{(N''_{Pra})^{1/3}} = \frac{11.593}{(262)^{1/3}} = 1.8142$$

$$\frac{N_{Nuoc}}{(N''_{Rea})^{2/3} (N''_{Pra})^{1/3}} = \frac{11.593}{(379)^{2/3} (262)^{1/3}} = 0.034889$$

$$\frac{N_{Nuoc}}{(N''_{Pra})^{1/3} \left(\frac{D_a}{D_c}\right)^{0.1}} = \frac{11.593}{(262)^{1/3} (7.5/34)^{0.1}} = 2.171$$

Heat Transfer to the Fluids Flowing Inside the coil

Table 24, Run no.1

Bulk temperature of the fluid entering the coil, $T_{ci} = 41.85^{\circ}\text{C}$

Bulk temperature of the fluid at the exit end of entering the coil, $T_{co} = 42.7^{\circ}\text{C}$

Mass rate of flow of the fluid flowing through the coil, $W_c = 37 \text{ kg/min}$

Temperature of the fluid in the agitated vessel, $T = 56.32^{\circ}\text{C}$

Using the same procedure as described previously

$$(\Delta t_c)_m = T - \frac{T_{ci} + T_{co}}{2} = 56.32 - \frac{42.7 + 41.85}{2} = 14.045^{\circ}\text{C}$$

$$U_{oc} = \frac{Q_c}{A_c \times (\Delta t_c)_m} = \frac{W_c (T_{co} - T_{ci}) \times c_p}{A_c \times (\Delta t_c)_m}$$

$$\frac{37 \times (42.7 - 41.85) \times 1 \times 60}{0.259 \times 14.045} = 518.74 \text{ Kcal/hrm}^2 \text{ }^{\circ}\text{C}$$

From the Wilson plot (Fig. 22) for aqueous 2% CMC-A the intercept has been obtained as 0.000905

$$\frac{1}{518.74} = \frac{1}{h_{oc}} + 0.000905$$

Or

$$h_{oc} = 976.562 \text{ Kcal/hrm}^2\text{0c}$$

Liquid film coefficient inside the coil : h_{ic}

$$\frac{1}{h_{ic}} = \left(\frac{1}{U_{oc}} - \frac{1}{h_{oc}} \right) \frac{D_a}{D_o} - \left(\frac{\Delta x_c}{k_m} \times \frac{D_t}{D_m} \right) = \left(\frac{1}{518.74} - \frac{1}{976.562} \right) \frac{1.898}{2.25} - \frac{0.176 \times 1.898}{340 \times 2.074}$$

$$\frac{0.8435}{518.74} - \frac{0.8435}{0.001275} - 0.0000047$$

Or

$$h_{ic} = 1853.9 \frac{\text{Kcal}}{\text{hr}} \text{m}^2\text{0c}$$

Nusselt Number : N_{Nuic} (Experimental)

$$N_{Nuic} \text{ (Experimental)} = \frac{h_{ic} \times D_t}{k}$$

$$\frac{1853.9 \times 1.898}{15.1 \times 36000 \times 10^{-4}} = 64.728$$

Velocity of the fluid flowing through the coil : U

$$U = \frac{W_c}{A \times \rho}$$

Area of cross section of the coil tube : A

$$A = \frac{\pi}{4} D_t^2$$

$$U = \frac{37 \times 1000}{60} \times \frac{\pi}{4} \times (1.898)^2 \times 1.02 = 213.45 \text{ cm/sec}$$

Consistency Index : k

From fig. 7 the value of k at temperature T_{cav} , i.e., at 42.275°C is obtained to be 0.89.

Pressure drop : ΔP

The pressure drop measurements were also done by a manometer during the heat transfer process and at the present prevailing conditions, using mercury manometer, the difference in the height of the two limbs Δh is obtained equal to 30.7 cm

$$\Delta P = \Delta h (\rho_{hg} - \rho) g / g_c = 30.7 (13.6 - 1.02)$$

Shear Stress : τ_w

$$\tau_w = \frac{D_t \times \Delta P}{4L} = \frac{1.898}{4 \times 366.48} \times 30.7 (13.6 - 1.02) = 0.49998 \text{ gm(f)/cm}^2$$

Viscosity : μ_2 (defined by equation III. 24)

$$\mu_2 = k' \left(\frac{\tau_w x g_c}{k'} \right)^{(n-1)/n} = \left(\frac{(3 x 0.793)+1}{4 x 0.793} \right)^{0.793} x 0.89 \left[\frac{0.49998 x 981}{\left(\frac{(3 x 0.793)+1}{4 x 0.793} \right)^{0.793} x 0.80} \right]^{\frac{0.793-1}{0.793}}$$

$$= 0.17088 \text{ gm/sec cm}$$

Differential viscosity : μ_d (defined by equation III.34)

$$\mu_d = \frac{4n^2}{3n+1} x \mu_2 = \frac{(0.793)^2}{(3 x 0.793) + 1} x 0.17088$$

$$= 0.1355078 \text{ gm/sec cm}$$

Reynolds number : N_{Re_2}

$$N_{Re_2} = D_t \rho \frac{U}{\mu_2} = \frac{1.898 x 1.02 x 213.45}{0.17088} = 2.39 x 10^3$$

Prandtl Number : N_{Pr_2}

$$N_{Pr_2} = \frac{c_p \mu_2}{k} = \frac{1 x 0.17088}{15.1 x 10^{-4}} = 113.16$$

Graetz Number : N_{GZ}

$$N_{GZ} = \frac{W_c x c_p}{k x L} = \frac{37 x 1000 x 1.0}{60 x 15.1 x 10^{-4} x 366.48} = 1114.35$$

Nusselt Number : N_{Nuis} , calculated from relationship presented by Pigford (27), for laminar heat transfer in straight pipe

$$N_{Nuis} = 1.75 \left(\frac{3n+1}{4n} \right)^{1/3} N_{GZ}^{1/3}$$

$$= 1.75 x \left(\frac{3x0.793+1}{4x0.793} \right)^{1/3} x (1114.35)^{1/3} = 18.587$$

Dean Number : N_{D_2}

$$N_{D_2} = N_{Re_2} \left(\frac{D_t}{D_c} \right)^{1/2} = 2.39 x 10^3 \left(\frac{1.898}{34.0} \right)^{1/2} = 564.6$$

Calculated value of N_{Nuic} from the present correlation (equation III.26)

$$N_{Nuic} (\text{calculated}) = \left(1 + 0.066 N_{D_2}^{1/2} N_{Pr_2}^{0.12} \right) N_{Nuis}$$

$$= 1 + 0.066 x (564.6)^{1/2} x (113.16)^{0.12} x 18.587 = 69.95$$

$$\frac{N_{Nuic}}{N_{Nuis}} = \frac{64.728}{18.587} = 3.482$$

$$\left(\frac{N_{Nuic}}{N_{nuis}} - 1\right) N_{D_2}^{-1/2} = (3.482 - 1) (564.6)^{-1/2} = \frac{2.482}{23.75}$$

$$\left(\frac{N_{Nuic}}{N_{nuis}} - 1\right) N_{Pr_2}^{-0.12} = \frac{2.482}{1.763} = 1.407$$

$$\frac{N_{Nuic}}{(N_{Re_2})^{2/3}} = \frac{64.728}{(2.39 \times 10^3)^{2/3}} = 0.36364$$

$$\frac{N_{Nuic}}{(N_{Pr_2})^{1/3}} = \frac{64.728}{(113.16)^{1/3}} = 13.4$$



Research Paper

LAMINAR FORCED CONVECTION TO FLUIDS IN
COILED PIPE SUBMERGED IN AGITATED VESSELAnsar Ali S K^{1*}, L P Singh² and S N Gupta³*Corresponding Author: Ansar Ali S K, ✉ ansnil@gmail.com

Laminar forced convection in curved pipe placed in agitated vessel has been studied experimentally for the flow of both Newtonian fluid (water) and power law non-Newtonian fluids (5%, 1% and 2% aqueous CMC solutions). The heat transfer has been calculated using modified Wilson method. It is found that the Nusselt number is function of Dean and Prandtl numbers. The following correlation is found for flow of Newtonian and power law non-Newtonian fluids flowing through helical coils.

$$\frac{N_{uic}}{N_{uis}} = 1 | 0.066 N_{D2}^{1/2} N_{Re}^{0.12}$$

Keywords: Agitated vessel, Newtonian, Non-newtonian power law, Wilson method

INTRODUCTION

Laminar forced convection in curved pipes has received considerable attention. From the work of Kubair and Kuloor (1966), Rajshekharan *et al.* (1966 and 1970), Akiyama and Cheng (1971) and Dravid *et al.* (1971), it is found that not much work has been done for non-Newtonian fluids in curved pipes. The existing heat transfer data in the literature for fully developed laminar forced convection in curved pipe with uniform wall temperature are rather limited and incomplete. For Newtonian fluids, perturbation method was applied by

Maekawa for extremely low Dean Number. The boundary layer approximation near the wall was presented by Akiyama and Cheng (1971) for high dean number of order one. Dravid *et al.* (1971) presented the numerical results in the thermal entrance region for Dean Number less than 225 and $N_{Pr} = 5$. The improved later work of Akiyama and Cheng (1972) shows that the ratio of heat transfer coefficient in coil and in straight pipe is a function of Dean Number as well as Prandtl number.

Thus there is a possibility of obtaining a suitable correlation of the following form

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$$\frac{N_{Nuis}}{N_{Nuis}} - 1 = \phi_1(N_D, N_{Pr}) \quad \dots(1)$$

The advantage of the above form of correlation is that for small N_D , the Equation (1) satisfies the condition that $N_{Nuis} = N_{Nuis}$ for very small N_D . N_{Nuis} may be calculated for isothermal laminar heat transfer in straight tube for the case of uniform wall temperature and the parabolic velocity distribution by following correlation as suggested by Mujawar and Raja Rao (1997).

$$N_{Nuis} = 1.75(N_{Gz})^{1/3} \quad \dots(2)$$

For non-Newtonian fluids the above equation takes the form

$$N_{Nuis} = 1.75 \left(\frac{3n+1}{4n} \right) (N_{Gz})^{1/3} \quad \dots(3)$$

Here again it is important to note that dean number contain Reynolds number and the viscosity term appears in both Reynolds and Prandtl numbers. It is suggested that the effective viscosity at the shear stress prevailing at the wall should be used for evaluating the Reynolds and Prandtl numbers. From the above discussion it may be concluded that the correlation may be written as follows

$$\frac{N_{Nuis}}{N_{Nuis}} - 1 = C_3 N_{D2}^{b5} N_{Pr}^{b6} \quad \dots(4)$$

The fluid heat transfer coefficient h_{oc} were calculated from overall coefficient U_{oc} using Wilson graphical method. $\frac{1}{U_{oc}}$ versus $\frac{1}{N^{2/3}}$ plot for water gives y_c equal to 0.00012. For different combinations of aqueous CMC solutions in the agitated vessel and flowing

through the coils $\frac{1}{U_{oc}}$ is plotted against $\frac{1}{N^{(3-n)/3}}$

yielded separate straight lines for different blade diameters. The respective y_c values were found to be 0.000445, 0.000575 and 0.000905 respectively.

The outside heat transfer coefficient of the coil tube from the coil side over all heat transfer coefficient U_{oe} was calculated by following relation.

$$\frac{1}{U_{oc}} = \frac{1}{h_{oc}} + Y_c \quad \dots(5)$$

where

$$Y_c = \frac{1}{h_{ic}} \left(\frac{D_c}{D_t} \right) + \frac{\Delta_x D_c}{K_m D_{tm}} + \text{dirt resistance} \quad \dots(6)$$

and

$$h_{oc} = Y_c N^{(3-n/3)} \quad \dots(7)$$

The calculated values of $\frac{1}{U_{oc}}$ and h_{oc} for 0.5% CMC, 1% CMC, 2% CMC that of 4% CMC solution and that of water for blade diameter 7.5 cm, 12.7 cm and 18.35 cm have been used in the present work.

The liquid film coefficient inside the coiled tube, i.e., h_{ic} was calculated by the relation

$$\frac{1}{h_{ic}} = \left(\frac{1}{U_{oc}} - \frac{1}{h_{oc}} \right) \frac{D_t}{D_o} \quad \dots(8)$$

For a set of observation the fluid flow rate in the coil was varied at a constant rotational speed of the agitator and for that set of readings h_{oc} was maintained constant. The value of h_{oc} was obtained from the Wilson plot according to the relation

$$\frac{1}{h_{oc}} = \frac{1}{U_{oc}} - Y_c \quad \dots(9)$$

EXPERIMENTAL PROCEDURE

The experimental setup consisted of a flat bottomed cylindrical test vessel of 45.25 cm inner diameter and 60 cm. heights made from 1/8 inch thick copper sheet. The vessel was jacketed by providing annular space surrounding it with another cylinder of 44 cm height and made from 1/8 inch thick G.I sheet. The jacket and vessel assembly was adequately insulated from outside. A helical coil having 34 cm mean coil diameter and made from 1.898 cm internal diameter and 2.25 cm outer diameter copper tubing was placed in the centre of the test vessel.

A rectangular tank of about 300 litre capacity filled with heaters was used to heat the water to a pre-determined temperature which in turn was circulated through the jacket (annular space around the test vessel) with the help of a centrifugal pump.

The fluid in the test vessel was agitated by marine type agitator fitted in the centre of the coil. The agitator shaft was driven at a known speed by a 2HP electric motor through a reduction gear assembly. Provision was made for replacement of impeller of desired shape and size.

The water in the hot water tank was heated and the temperature of fluid was brought to the desired level. The temperature was controlled to a pre determined value with the help of temperature controller. The water circulation was then started in the coil as well as in the jacket and adjusted by means of regulating valves and bypasses. The agitator was then started at a fixed rpm.

At steady state condition the inlet and outlet water temperatures in the jacket, mass rate of

water flow through jacket and that of fluid in the coil, rpm of the agitator and temperatures of fluid in the agitated vessel and that of water in the storage tank were noted. The test fluid side wall temperatures of the test vessel were noted at different locations and at different heights from its bottom with the help of copper constantan thermocouples.

The readings were duplicated to ensure the steady state and to eliminate any error in measurement. Similar measurements were made by varying the flow rate in the coil and then by varying the rotational speed of the agitator. The above procedure was repeated for all the fluids.

The heat transfer characteristics of five fluids, viz., water and four aqueous CMC solutions of concentration 0.5, 1, 2 and 4% by weight have been investigated. The rheological properties were determined with the help of capillary tube viscometer. All the CMC solutions were found to be pseudoplastic in nature obeying power law relation.

The flow behavior indices were found to be 0.937, 0.851, 0.793, and 0.698 for 0.5%, 1%, 2% and 4% aqueous CMC solutions respectively. Thermal conductivity of CMC solutions were determined by comparative concentric cylinders and were found to be equal to that of water. Specific heats of the solutions, as measured by calorimetric method, were found to be nearly equal to that of water.

RESULTS AND DISCUSSION

In order to obtain exponent b_5 and b_6 and constant C_3 in Equation (4) the proposed form of correlation for laminar forced convection to non-Newtonian fluids in helical coils 0.5, 1 and

2% CMC solutions were investigated in a helical coil of 34 cm diameter made from a tube of diameter 1.898 cm. Before conducting the heat transfer runs, pressure drop measurements were made to verify the fitness of the coil. Coil flow is compared with capillary shear flow in Figure 1. Friction factor f_c is shown in Figure 2 against Reynolds number N_{Re_2} defined by the following equation.

$$N_{Re_2} = \frac{D_t U_p}{\sim 2} \dots(10)$$

Figure 1: Flow Diagram for 0.5% , 1% and 2% CMCA Through HT Test Coil

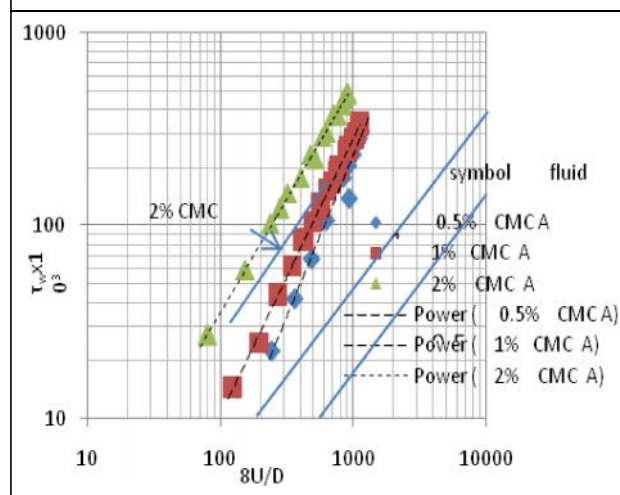


Figure 2: Variation of Friction Factor with Reynolds Number for Water, 2% , 1% and 0.5% CMCA Through HT Test Coil

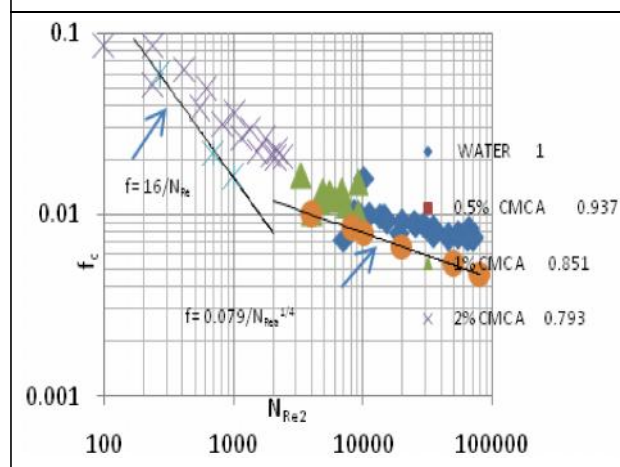
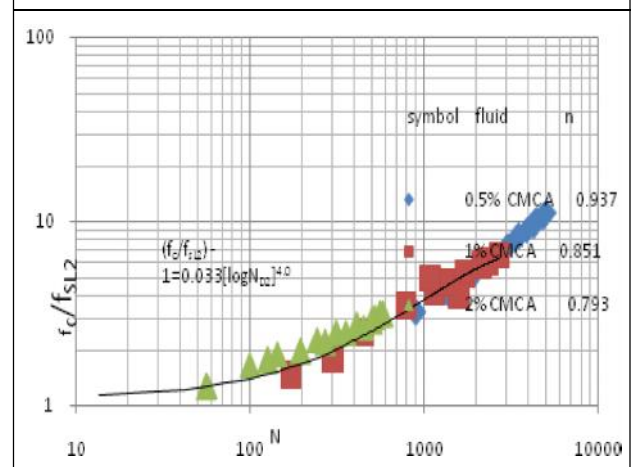


Figure 3: Laminar Flow Friction Factor in Helical Coils as a Function of Dean Number for Non-Newtonian Fluids



The ratio of friction factor, f_c , for coil to that of straight pipe, f_{SL} evaluated at Reynolds number, N_{Re_2} , from equation $f_{SL} = 16/N_{Re_2}$ is plotted against N_{D_2} in Figure 3.

$$\left(\frac{f_c}{f_{sl}}\right) - 1 = 0.33(\log N_{D_2})^{4.0} \dots(11)$$

The Equation (11) proposed by Berger *et al.* (1983) is also shown in this figure. The excellent agreement of the data with above equation indicates the fitness of the coil and the heat transfer data obtained in this coil could be placed for its accuracy.

Three non-Newtonian fluids: 0.5% ($n = 0.937$), 1% ($n = 0.851$) and 2% ($n = 0.793$) CMC in water, representative of pseudo plastic fluids obeying power law, and water, a Newtonian fluids were investigated. Most of the data, obtained, were in laminar region and only a few non-Newtonian data could fall in turbulent region.

Om Prakash *et al.* have shown that the fluid flow data in helical coils over wide range of coil diameter and fluid behavior can

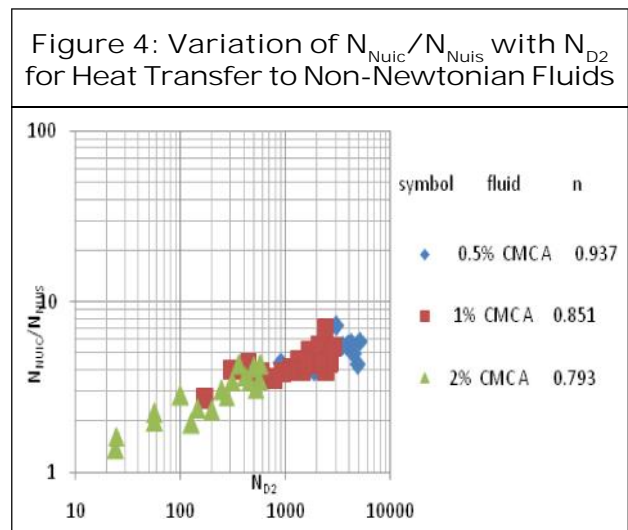
successfully be correlated by the use of apparent viscosity μ_z evaluated at the wall shear stress in Reynolds number N_{Re_2} . This suggests the possibility of analogous

correlation in terms of the ratio $\left(\frac{N_{Nuic}}{N_{Nuis}}\right)$. N_{Nuis} was evaluated by Equation (3).

Effect of Dean Number

In order to find the effect on the ratio of heat transfer in coil to heat transfer in straight pipe,

the first attempt was to plot the ratio $\left(\frac{N_{Nuic}}{N_{Nuis}}\right)$ against N_{D_2} on a logarithmic scale as shown in Figure 4.



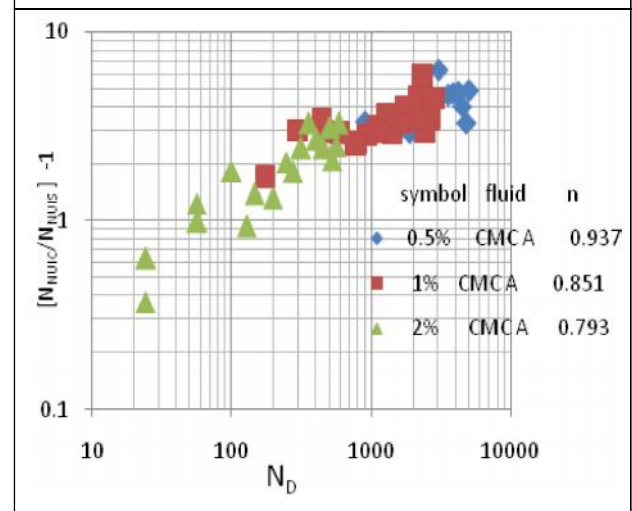
No direct relationship is seen between the two variables in this manner. At very low Dean Number or very small curvature, effect of secondary flow will be negligible and then N_{Nuic} will be equal to N_{Nuis} . Therefore, the next

attempt was to plot $\left(\frac{N_{Nuic}}{N_{Nuis}}\right) - 1$ against N_{D_2} .

This plot is shown in Figure 5.

No net conclusion could be drawn even from this figure except for some qualitative

Figure 5: Variation of $\left[\left(\frac{N_{Nuic}}{N_{Nuis}}\right) - 1\right]$ with N_{D_2} for Heat Transfer to Non-Newtonian Fluids



ones. The heat transfer coefficient in coil is always higher than in straight pipe as it is seen

that $\frac{N_{Nuic}}{N_{Nuis}}$ is always greater than unity. The ratio

h_c/h_{is} increases with Dean number, N_{D_2} . These data do not follow any particular trend. This may be explained by the fact that due to non Newtonian behavior Prandtl number changes by change of flow rate affecting the heat transfer rate. Data plotted for three fluids in Figure 5 have wide range of Prandtl number. However, an exponent of Dean Number $\frac{1}{2}$ correlated the data much better. This trial has been made on the basis of theoretical results presented by previous investigators.

Effect of Prandtl Number

Prandtl number N_{pr2} was calculated by using apparent viscosity, $\tilde{\mu}_2$ evaluated at wall shear stress, τ_w , obtained from the measured pressure drop. The $\tilde{\mu}_2$ may be evaluated by the following derived expression

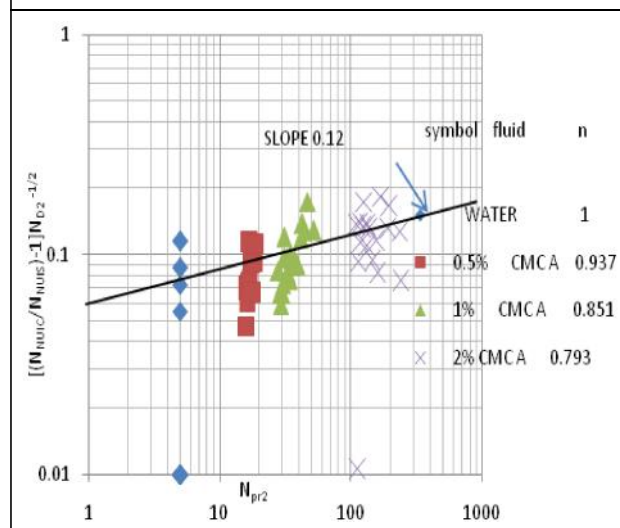
$$\tilde{\mu}_2 = k'(\tau_w / k')^{\frac{n-1}{n}}$$

Subsequently $N_{Re_2} = DtU_{\infty}/\mu_2$ and

$$N_{D_2} = N_{Re_2} (D/D_c)^{1/2}$$

In order to understand the effect of N_{pr2} on the ratio h_{ic}/h_{is} , a logarithmic plot of $\left(\frac{N_{uic}}{N_{uis}}\right)^{-1} N_D^{-1/2}$ versus N_{pr2} is shown in Figure 6. Figure 6 clearly shows the effect of N_{pr2} . Many other trials were made by changing the exponent of Dean number N_{D_2} , in the ordinate variable of Figure 6.

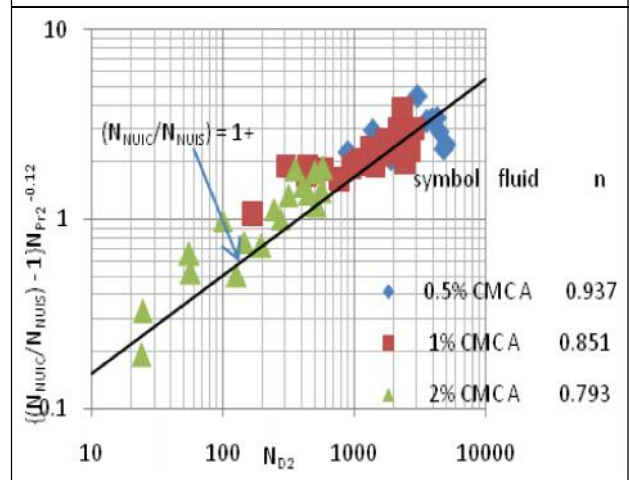
Figure 6: Laminar Flow Heat Transfer Correlation for Non-Newtonian Fluids Through Coils: Effect of Prandtl Number



Values of exponent other than $\frac{1}{2}$ on N_{D_2} tend to increase the scatter of data points. The slope of mean line shown represents the exponent of N_{pr2} and was estimated to be 0.12 for Prandtl number range of 4.9 to 225. Finally

$\left(\frac{N_{uic}}{N_{uis}} - 1\right) N_{pr2}^{-0.12}$ is plotted against N_{D_2} in Figure 7 to verify the exponent of N_{D_2} . The mean line passing through data points show an exponent equal to $\frac{1}{2}$. The resulting correlation is obtained as:

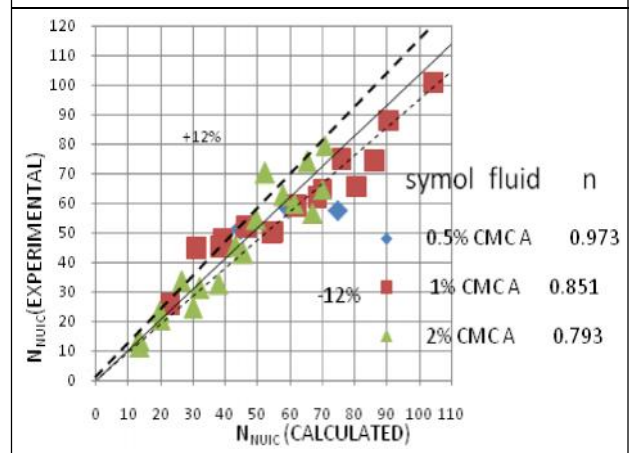
Figure 7: Laminar Flow Heat Transfer Correlation for Non-Newtonian Fluids Through Coils



$$\frac{N_{uic}}{N_{uis}} = 1 + 0.066 N_{D_2}^{1/2} N_{pr2}^{0.12} \quad \dots(12)$$

Figure 8 compares the experimental Nusselt number with corresponding values calculated from Equation (12). It is found that that laminar flow heat transfer data in the range $24 < N_D < 2000$, $N_{pr2} < 225$ and $0.793 < n \leq 1$ can successfully be correlated using Equation (12) with a standard deviation of 15.5%.

Figure 8: Comparison Between Experimental and Calculated Values of Laminar Nusselt Number N_{NUIIC} (Heat Transfer to Non-Newtonian Fluids Through Coil)



According to Ozisik and Topa kaglu's theoretical approach (08), there is possibly of two regions to exist, namely the heated and cooled region in the fluid space and that too depends upon Prandtl number.

This equation is similar to the equation suggested by Mori and Nakjayama (09) for flow of Newtonian fluids across tube banks

where the ratio $\frac{N_{uic}}{N_{uis}}$ is the function of N_{pr2} and N_{D_2} .

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APPENDIX

Nomenclature

b_5, b_6	constant in Equations (4)
C_3	constant in Equation (4)
D_c	diameter of the coil helix, cm
D_t	inner diameter of the straight or coil tube
D_o	outer diameter of the straight or coil tube
f_{sL}	laminar flow friction factor in straight pipe
f_c	friction factor in coil
h_{ic}	coil inside heat transfer coefficient, kcal/hr m ² °C
h_{oc}	coil outside heat transfer coefficient, kcal/hr m ² °C
k'	consistency index, gm sec n ⁻² /cm
N	speed (r.p.m)
N_{pr}	Prandtl number
N_{NUIC}	Nusselt number, $h_{ic}D_t/k$
N_{NUIS}	Nusselt number, $h_{is}D_t/k$
N_D	Dean number
N_{GZ}	Graetz number, wc_p/kL
N	speed (r.p.m)
n'	generalized flow behavior index
U_{OC}	coil overall heat transfer coefficient, Kcal/hr m ² °C
h_{oc}	coil outside heat transfer coefficient, Kcal/hr m ² °C
h_{ic}	coil inside heat transfer coefficient, Kcal/hr m ² °C
Y_c	resistance of the jacket side fluid and the wall
Δ_x	width of the agitated vessel wall, cm
N_{Re_2}	Reynolds number defined by Equation (10)
U	average velocity, cm
P	density, gm/cm ³
$\tilde{\nu}_2$	effective or pseudo shear viscosity, gm/cm
\dagger_w	shear stress, gm(f)/cm ²
$\tilde{\nu}_e$	effective viscosity
$\tilde{\nu}_a$	apparent viscosity at the impeller tip



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Heat Transfer to Newtonian and Non-Newtonian Fluids in mechanically Agitated Vessel

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Abstract: Heat transfer to Newtonian (water) and power law Non-Newtonian (1%, 2% and 4% aqueous CMC solution) fluids in the agitated vessel is investigated. The data have been obtained for fluid agitated by three marine agitators of 7.5, 12.7 and 18.35 cm diameter respectively. The heat transfer coefficient has been calculated using Wilson graphical method with modification suggested by Om Prakash et al. The heat transfer data for agitated water and 1, 2 and 4% aqueous CMC solutions for impeller diameters have been correlated by the equation $N_{Nu} = 0.302 N_{Re}^{0.42} N_{Pr}^{1/3} \left(\frac{D}{D_i}\right)^{0.1}$ with standard deviation of 8.03%.

Keywords: Agitator, Impeller, Non-Newtonian, Power Law Fluid, Power Law.

I. INTRODUCTION

From the available literature it is seen that the average heat transfer coefficient depends on the geometrical and physical factors. The correlation between them is not so simple because of complex relationship between large numbers of variables involved. The heat transfer rate in an agitated vessel, either from the vessel wall or from the wall of coils, depends upon velocity and temperature profiles near the wall. The velocity profile itself, which affects the temperature profile, depends on geometrical factors, such as the impeller diameters and its nature, position and speed, the diameter of the fluid reservoir, the diameter and pitch of the coils, the diameter of the coil tube, the fluid height in the tank and also on the position and dimensions of the baffles. The physical parameters are the physical properties of the fluids, specially, the velocity, the thermal conductivity, density, specific heat and their variation with the temperature. Physical parameters are normally grouped in terms of Prandtl number, Reynolds number and viscosity correction factors whereas geometrical properties are considered in Reynolds number and various geometric dimensionless groups.

The non-linear relationship between shear stress and shear rate for non-Newtonian materials makes the development of the heat transfer correlation more complex. For flow of such materials difficulties arise in specifying the Reynolds number, Prandtl number and viscosity ratio. Therefore an appropriate choice of viscosity is necessary to obtain suitable correlation for heat transfer. Carreau, Charest and Corneillo (1) studied heat transfer to pseudoplastic fluids in jacketed vessel using turbine agitator and employed a generalized Reynolds number analogous to that of laminar flow in pipes

while Reynolds number was defined by using differential viscosity at high shear rate. The flow behavior of the fluids in agitated vessel is not correctly and adequately described by the Reynolds number used by them. Thus some variation exists in the final correlation in the coefficients for Newtonian and non-Newtonian fluids. Gluz and Pevlushenko (2) used a viscosity based on average shear rate corresponding to shear rate at the surface of a rotating cylinder, and the Reynolds number defined by Gluz et al. describes the flow behavior in a better manner compared to that used by Carreau et al.

Hagedorn and Salamone (3) considered the flow pattern at the vessel wall for power law fluids and presented the correlation on the basis of dimensional analysis of basic equations. OC Sandall et al (7) also considered the shear rate at the wall and correlated the data on Newtonian and non-Newtonian fluids for anchor agitator assuming shear rate proportional to ratio of agitator tip speed and the clearance between the agitator tip and the wall. Many other workers (5,6,7,8,9,10,11,12) have made efforts to correlate the similar data by using average shear rate derived from agitator power measurements following the methods of Metzner and Otto (13) and Calderbank and Moo Young (14,15). For power law fluids, shear stress and shear rate relationship at the rotor of the bob viscometer in an infinite fluid (considered as the surface of the cylinder rotating in infinite fluid) is given by

$$\tau_s = \left[r \frac{d\left(\frac{u_\theta}{r}\right)}{dr} \right]_s^n = K(4\pi N/r)^n \quad (1)$$

and the relationship for generalized power law may be stated as

$$\tau_s = K''(4\pi N)^{n''} \tag{2}$$

$$\left[r \frac{d(u_{\theta}/r)}{dr} \right]_s = \frac{4\pi N}{n''} \tag{3}$$

$$K'' = K/(n)^n \tag{4}$$

And
For power law fluids ($n'' = n$) various viscosity and Reynolds number expressions may be defined as follows Pseudo shear viscosity

$$\mu_e'' = \tau_s/4\pi N = \frac{K}{n^n} (4\pi N)^{n-1} \tag{5}$$

and the corresponding Reynolds number

$$N_{Re}'' = D_a U_s \rho / \mu_e = \frac{(n)^n D_a^2 N^{2-n} \rho}{K(4)^{n-1} \pi^{n-2}} \tag{6}$$

Apparent or shear viscosity may be written as

$$\mu_a'' = \frac{\tau_s}{\left[r \frac{d(u_{\theta}/r)}{dr} \right]_s} = n^{(1-n)} K(4\pi N)^{n-1} \tag{7}$$

$$N_{Rea}'' = \frac{D_a^2 N^{2-n} \rho}{n^{1-n} K(4)^{n-1} (\pi)^{n-2}} \tag{8}$$

and
Differential viscosity

$$\mu_d'' = \frac{d\tau_s}{d \left[r \frac{d(u_{\theta}/r)}{dr} \right]_s} \tag{9}$$

or

$$\mu_d'' = \frac{\tau_s}{r \left[\frac{d(u_{\theta}/r)}{dr} \right]_s} \frac{d \ln \tau_s}{d \ln \left[r \frac{d(u_{\theta}/r)}{dr} \right]_s} = n \mu_a'' \tag{10}$$

From equations (5), (7) and (9) one obtains

$$\mu_d'' = n \mu_a'' = n^2 \mu_e'' \tag{11}$$

and relationship between various Reynolds numbers N_{Re}'' , N_{Rea}'' and N_{Red}'' on the basis of pseudo shear, shear and differential viscosity, respectively, may be written as

$$N_{Re}'' = n N_{Rea}'' = n^2 N_{Red}'' \tag{12}$$

Similarly corresponding Prandtl numbers may be defined as

$$N_{Prd}'' = n N_{Pra}'' = n^2 N_{Pr}'' \tag{13}$$

$$N_{Pr}'' = \frac{c_p K(4\pi N)^{n-1}}{(n)^n k}$$

Where

The analysis of the flow pattern around the mixing impeller is not so simple. A compromise between experimental and physical pictures of the flow pattern is needed to obtain suitable parameters for the correlation. Let us consider the case of flat plate submerged in a fluid flowing in a laminar fashion. If the thickness of the flat plate is small compared to width, most of the drag on the plate will be due to pressure difference between front and rear of the plate which is caused by the motion of the fluid near the surface. Shear stress causing any drag will be negligible. Now let us consider the

flow situations around the flat blade of turbine or paddle rotating in a fluid. At low speed of rotation the flow pattern may be considered similar to that of flat plate submerged in a fluid, in motion. At higher speed, with more number of blades the extent of the separation of flow will decrease and the viscous energy dissipation becomes the controlling factor. Thus the use of dimensionless groups defined by visualizing the system as a cylinder of diameter equal to that of blade rotating in an infinite fluid medium will be better than compared to the paradoxical definitions of Reynolds number and viscosity analogous to pipe flow.

For heat transfer from the wall of the vessel agitated by a turbine agitator, the fluid flow along the wall of the vessel is upward and the resistance to the heat transfer is mainly controlled by the viscous sub-layer along the wall, with a large velocity and temperature gradient in radial direction. The heat transferred through the viscous sub layer is completely mixed with the bulk of fluid between the sub layer and the impeller. Under steady state condition the momentum, mass and energy equations for power law fluids in the region near the wall may be simplified and presented in the following form:

$$\rho u_r \left(\frac{\partial u_z}{\partial r} \right) = - \frac{\partial p}{\partial z} - \frac{1}{r} \frac{\partial}{\partial r} \left[-rk \left(\frac{\partial u_z}{\partial r} \right)^n \right] + \rho g_z \tag{14}$$

$$\frac{1}{r} \frac{\partial}{\partial r} (\rho r u_r) = 0 \tag{15}$$

$$u_r \frac{\partial T_1}{\partial r} = \frac{K_t}{\rho c_p} \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T_1}{\partial r} \right) \right] \tag{16}$$

The viscous sub layers thickness δ will be very small compared to the vessel diameter. As the Reynolds number, N_{Rea}'' increases the thickness of the viscous sub layer decreases. The δ will be a function of D_a and N_{Rea}'' and U_{rs} will be function of N_{Rea}'' and characteristic velocity, $\pi D_a N$.

The following dimensionless variables, now can be defined as

$$r^* = (r/D_a) N_{Rea}''^m \tag{17}$$

$$u_r^* = (u_r / \pi D_a N) N_{Rea}''^m \tag{18}$$

$$Z^* = \frac{z}{D_a}, \quad u_z^* = u_z / \pi D_a N \tag{19}$$

$$p^* = (p - p^0) / \rho (\pi D_a N)^2 \tag{20}$$

$$T^* = (T_i - T) / (T_s - T) \tag{21}$$

Substituting the above dimensionless variables in equations (14) and (16) it is observed that the velocity gradient $\frac{\partial u_z}{\partial r^*}$ is a function of $(N_{Rea}'')^{m(n+1)-1}$ and the temperature gradient $\delta T^* / \delta r^*$ is a function of $(N_{Rea}'')^{2m-1} / N_{Pra}''$.

Using Fourier law of heat conduction and Newton's law of cooling, the expression for total heat flux and heat transfer

Heat Transfer to Newtonian and Non-Newtonian Fluids in mechanically Agitated Vessel

coefficient can be obtained. The final expression for Nusselt number may be arranged.

$$N_{uj} = h_j D_T / K_t = D_T \frac{N_{Rea}^{n_m}}{H} \int_0^{H/D_a} \left(\frac{\partial T}{\partial r^*} \right)_w dz^* \quad (22)$$

Thus it is seen that N_{uj} is function of N_{Rea} , N_{Pra} and length ratio H/D_a . if non-isothermal correlation factor is included the correlation takes the form

$$N_u = C_1 N_{Rea}^{b_1} N_{Pra}^{b_2} (\mu_{ab} / \mu_{aw})^{b_3} \quad (23)$$

Retaining conventional values of Reynolds and Prandtl number indices $b_1=2/3$ and $b_2=1/3$ respectively and sieder tate correction factor $(\mu_{ab} / \mu_{aw})^{0.14}$, the equation 23 takes the form

$$N_{uj} = C_1 (N_{Rea})^{2/3} (N_{Pra})^{1/3} (\mu_{ab} / \mu_{aw})^{0.14} \quad (24)$$

Most of the data available for heat transfer in agitated vessel are for Newtonian fluids only. The similar data for non-Newtonian fluid is almost meager. Therefore experimental study was carried out to obtain data for heat transfer to purely viscous non-Newtonian fluids in agitated vessel.

II. EXPERIMENTAL

The experimental setup considered of a flat bottom test vessel of 45.25cm inner diameter and 60 cm. heights made from 1/8 inch thick copper sheet. The vessel was jacketed from 1/8 inch thick GI sheet. A rectangular tank of about 300 litre capacity filled with heaters was used to heat the water to a pre determined temperature which in turn was circulated through the jacket (annular space around the test vessel) with the help of a centrifugal pump. The fluid in the test vessel was agitated by marine type agitator fitted in the centre of the coil. The agitator shaft was driven at a known speed by a 2HP electric motor through a reduction gear assembly. Provision was made for replacement of impeller of desired shape and size. The water in the hot water tank was heated and the temperature of fluid was brought to the desired level. The temperature was controlled to a pre determined value with the help of temperature controller. The water circulation were then started in the jacket and adjusted by means of regulating valves and bypasses. The agitator was then started at a fixed rpm. At steady state condition the inlet and outlet water temperatures in the jacket, mass ratio of flow water from jacket test fluid from, rpm of the agitator and temperatures of fluid in the agitated vessel and that of water in the storage tank were noted.

The test fluid side wall temperatures of the test vessel were noted at different locations and at different heights from its bottom with the help of copper constantan thermocouples. The readings were duplicated to ensure the steady state and to eliminate any error in measurement. Similar measurements were made by varying the flow rate in the jacket and then by varying the rotational speed of the agitator. The above procedure was repeated for all the fluids. The heat transfer characteristics of five fluids, viz., water and four aqueous CMC solutions of concentration 0.5, 1, 0.2 and 4% by weight have been investigated. The rheological properties were determined with the help of capillary tube viscometer. All the

CMC solutions were found to be pseudoplastic in nature obeying power law relation. The flow behavior indices were found to be 0.937, 0.851, 0.793, and 0.698 for 0.5%, 1%, 2% and 4% aqueous CMC solutions respectively. Thermal conductivity of CMC solutions were determined by comparative concentric cylinders and were found to be equal to that of water. Specific heats of the solutions, as measured by calorimetric method, were found to be nearly equal to that of water.

III. RESULT AND DISCUSSION

In order to evaluate constant C_1 and Reynolds and Prandtl number indices b_1 and b_2 respectively the viscosity μ_a and following dimensionless groups were calculated.

$$N_{uj} = h_j D_T / K \quad (24)$$

Liquid film heat transfer coefficient h_j were calculated from over all coefficients U_j using Wilson graphical method modified by Om Prakash et al.

$$\mu_a^n = n^{1-n} (K) (4\pi N)^{n-1} \quad (25)$$

Reynolds and Prandtl number may be written as

$$N_{Rea}'' = \frac{D_a^n (\pi D_a N)^{2-n} \rho}{K n^{1-n} 4^{n-1}} \quad (26)$$

$$N_{Pra}'' = \frac{C_p K \left(\frac{4\pi N}{n} \right)^{n-1}}{K} \quad (27)$$

A. Effect of Prandtl Number

Prandtl number influence is shown in fig.1 where $N_{Nuj} / N_{Rea}^{n/3}$ is plotted against N_{Pra}'' and the influence of Reynolds number is eliminated by dividing N_{Nuj} values by $N_{Rea}^{n/3}$.

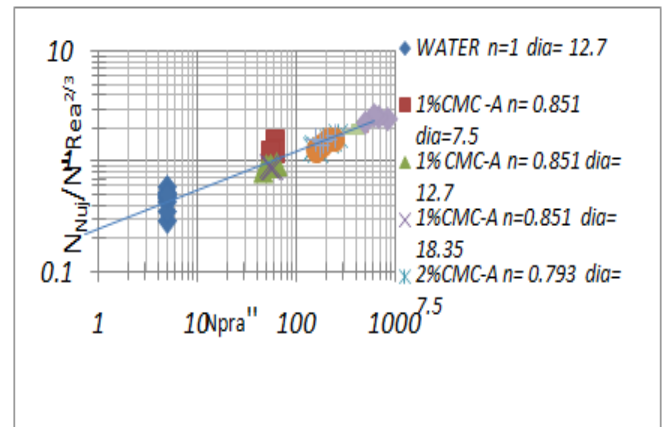


Fig.1. Jacket to agitated fluid heat transfer correlation (effect of prandtl number).

The exponent of N_{Rea}'' was chosen as $2/3$ from the previous experience. Prandtl number of polymeric solutions obeying power law varies with the speed of the agitator and is found to vary from a minimum of 4.9 for water to a maximum of 850 for 4% CMC solution in the present investigation. The average line through the data points of fig.2 gave the exponent b_2 of the Prandtl number equal to $1/3$ which agrees with Prandtl number exponent in heat transfer correlation for Newtonian fluids.

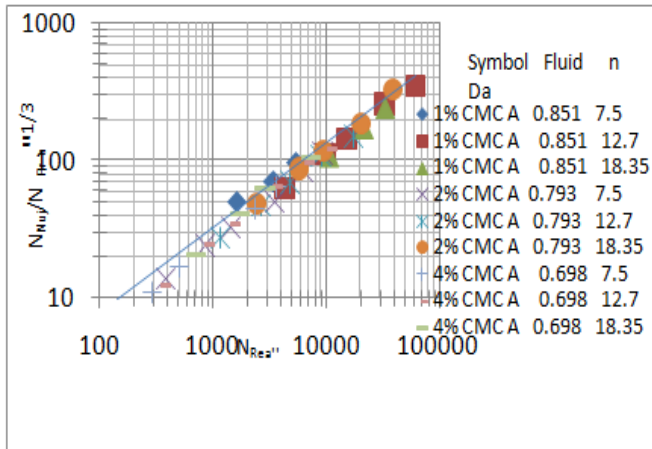


Fig.2. Heat transfer correlation for agitated non-Newtonian fluids (effect of Reynolds number).

B. Effect of Impeller Diameter

For studying the effect of impeller diameter, the heat transfer measurements were made with three impeller of diameter 7.5, 12.7 and 18.35 cm respectively. Although the diameter of impeller, D_a shows its effect is Reynolds number, an attempt has been made to find the effect of D_a/D_T ratio on Nusselt number. Fig.3 shows a correlation between $(N_{uj}/N_{Pra}^{1/3} N_{Rea}^{2/3})$ and D_a/D_T in which data of three fluids for D_a/D_T ratio 0.166, 0.282 and 0.403 have been plotted. Almost negligible effect of D_a/D_T is observed possibly because of very small range of D_a/D_T ratio covered in the present work.

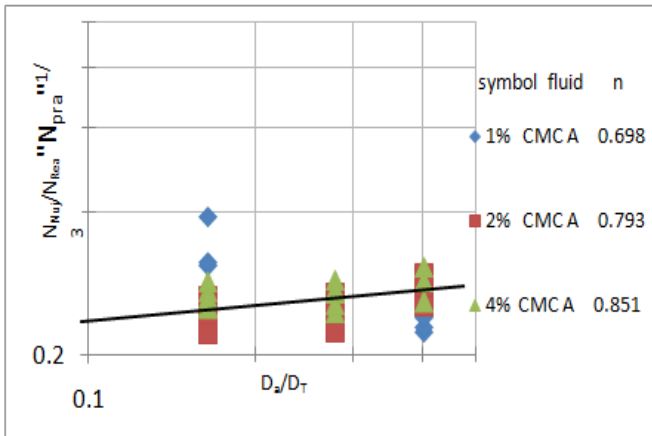


Fig.3. Heat transfer correlation for jacket to agitated fluids: effect of agitator diameter.

However, a mean line through data point shows the index of D_a/D_T equal to 0.1. Thus the overall effect of D_a , including its presence in Reynolds number, gives

$$h_j \propto D_a^{1.433} \tag{25}$$

Finally $N_{uj}/N_{Pra}^{1/3} (D_a/D_T)^{0.1}$ is plotted against Reynolds number N_{Rea} in fig.4 which includes data for water and 1,2 and 4% CMC solution. The method of least square gives the equation

$$N_{Nuj} = 0.302 N_{Rea}^{2/3} N_{Pra}^{1/3} \left(\frac{D_a}{D_T}\right)^{0.1} \tag{26}$$

It is interesting to note that all data of both Newtonian and non-Newtonian fluids are in excellent agreement with equation 26 for $290 < N_{Rea} < 1.4 \times 10^7$; $4.9 < N_{Pra} < 850$ and $0.166 < (D_a/D_T) < 0.403$.

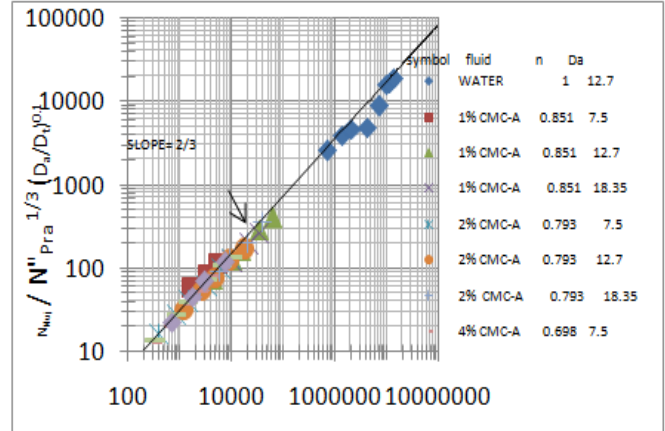


Fig.4. Heat transfer for jacket to agitated Newtonian and non-Newtonian fluids (effect of Reynolds number).

Fig.5 compares the experimental Nusselt numbers with those calculated from equation 26. The standard deviation of 50 data points from equation 26 is found to be 8.03%.

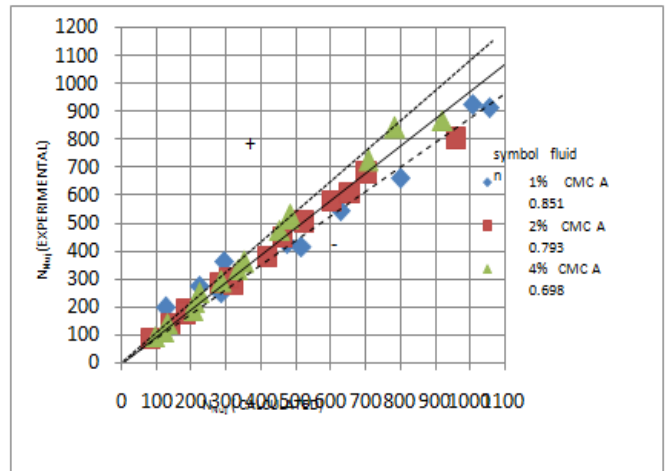


Fig.5. Comparison between experimental and calculated values of nusselt number nnuj (heat transfer from jacket to agitated fluids).

IV. CONCLUSION

- The Wilson plot of $1/u_j$ versus $Re_a^{2/3}$ is appropriate for the determination of individual heat transfer coefficient.
- For small temperature driving forces the non thermal correction is negligible.
- The heat transfer data for agitated Newtonian and non-Newtonian fluids have been successfully correlated by the viscosity of the fluid evaluated at the impeller tip having a cylinder of diameter equal to that of impeller rotating in an infinite fluid data of 1, 2 and 4% CMC for three impeller diameters have been correlated by the equation

$$N_{Nuj} = 0.302 N_{Rea}^{2/3} N_{Pra}^{1/3} \left(\frac{D_a}{D_T}\right)^{0.1}$$

Heat Transfer to Newtonian and Non-Newtonian Fluids in mechanically Agitated Vessel

With standard deviation of 8.03% for $290 < N''_{Rea} < 1.4 \times 10^7$; $4.9 < N''_{Pra} < 850$ and $0.166 < (D_a/D_T) < 0.403$. Using the above concepts of Reynolds and Prandtl numbers it is also possible to correlate the available published data for other non-Newtonian fluids obtained with different impeller geometries.

V. NOMENCLATURE

1. ${}^{\circ}C N_{Nu_j}$ Nusselt number, $h_j D_T/k$
2. N''_{Rea} Reynolds number defined by equation 8
3. N_{Pra} Prandtl number defined by equation
4. D_a agitator diameter, cm
5. D_T diameter of the agitated vessel
6. τ_s shear stress, gm(f)/cm²
7. r radial distance, cm
8. u_o outlet velocity
9. K consistency index, gm secⁿ⁻²/cm
10. N speed (r.p.m)
11. N flow behavior index
12. K'' consistency index, gm sec^{n''-2}/cm
13. n'' generalized flow behavior index
14. μ_e'' effective viscosity at the impeller tip defined by equation [11]
15. N_{Re}'' Reynolds number defined by equation [6]
16. U_s superficial velocity
17. P pressure, gm(f)/cm²
18. μ_e effective viscosity
19. μ_a'' apparent viscosity at the impeller tip defined by equation [13]
20. θ function of time, sec
21. μ_d differential viscosity, gm/cm sec
22. c_p specific heat, cal/gm⁰c
23. k thermal conductivity
24. N_{prd}'' prandtl number defined by equation [7]
25. k thermal conductivity, cal/cm sec⁰c
26. δ – boundary layer thickness, cm
27. T_i inlet temperature
28. T_s surface temperature
29. P^0 reference pressure, gm(f)/cm²
30. h_j heat transfer coefficient for jacketed vessel wall to fluid, Kcal/hr m² °C
31. H height of the fluid level in the vessel, cm
32. b_1, b_2, b_3 constants in equation [23]
33. μ_{ab} shear or apparent viscosity at bulk temperature
34. μ_{aw} shear or apparent viscosity at the wall temperature
35. c_1 constants in equations [23]
36. U_j jacket overall heat transfer coefficient, Kcal/hr m²

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